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GREAT LAKES GRAIN TRAFFIC.

BY R. J. DONOVAN.

Among those necessities of life furnished by the vegetable kingdom there are probably no more important products than those generally specified as grain, *i.e.*, wheat, corn, oats, etc. The growth of grain and its distribution to the various parts of the inhabited world have always demanded the services of a large percentage of those "who earn their bread by the sweat of their brow." The ship from ancient times has been the chief means of transportation of the grain crops, and we remember reading with what anxiety the hungry Romans awaited the coming of the grain-laden fleets from Africa, and have noted the frequent allusions to the same subject in the Bible.

Among those portions of the earth adapted to the growth of grain there are probably no sections so richly endowed by Nature for that purpose as the great rolling plains of the West and Northwest of our own country and Canada. These sections not only possess peculiar natural advantages in the soil, but they also border on the greatest natural highway of the world—the Great Lakes.

The remarkable growth and development of navigation and commerce on the Great Lakes is without parallel both in its rapidity and extent. Competition with railroads, particularly in the handling of bulk freight, such as grain, ore, coal, etc., has lately changed conditions previously existing. The altered condition of trade has very largely changed the type of vessel required, and has necessitated revisions of construction, so that at the present date there has been evolved a type of vessel peculiar to the lakes and eminently fitted for the purpose intended.

LIFTING BRIDGE AND GRAIN ELEVATORS AT BUFFALO, N. Y.

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Mechanical appliances at the various lake ports for the rapid handling of cargoes are, perhaps, unequaled in any part of the world, and the construction of the lake freighter must be such as to meet these modern and improved methods and conditions. So we may say that the greatest economy has been obtained not

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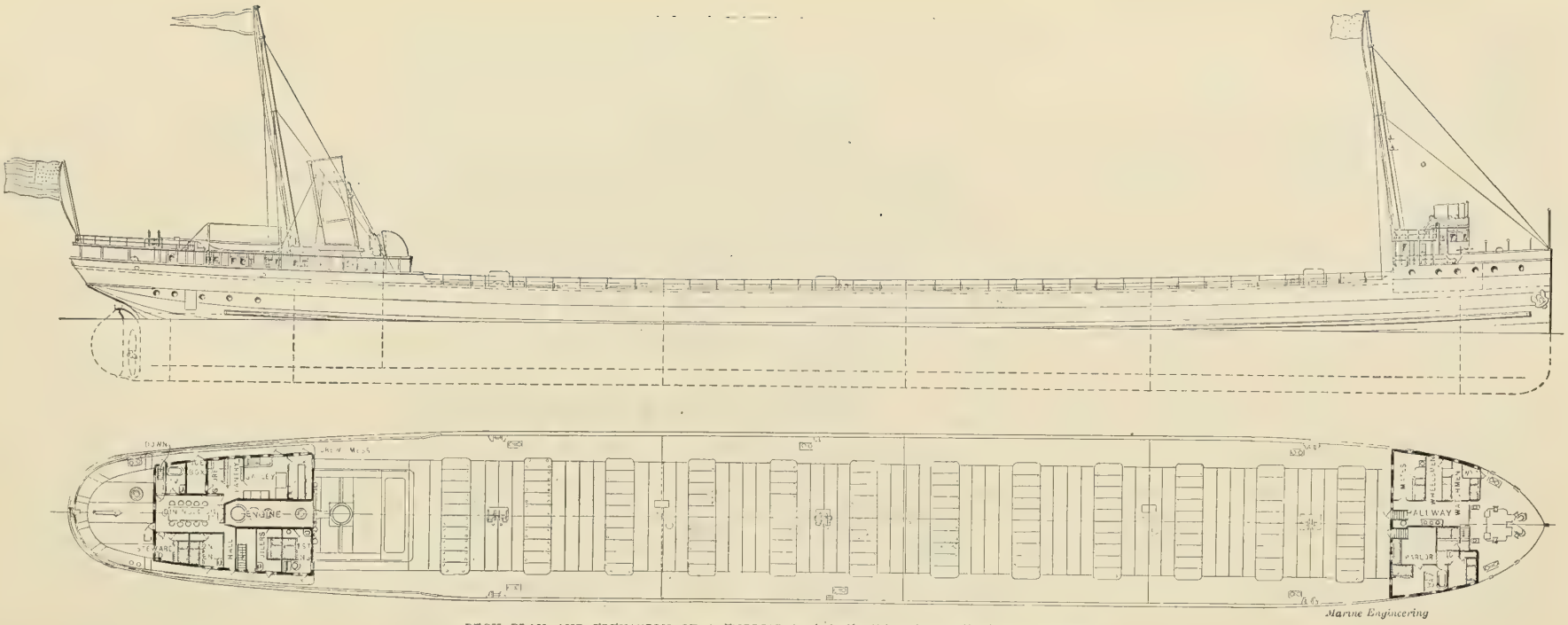


DECK PLAN AND ELEVATION OF A TYPICAL FREIGHT STEAMER OF THE GREAT LAKES.

only by the rapid terminal facilities, but also by the ability to transport, with the least operating cost, the greatest tonnage.

The purpose of this account is to give the reader a brief but accurate description of the Great Lakes grain traffic; involving a short account of the elevator, of the operations necessary in the handling of grain, and the vessels engaged in the trade. With a view to aid the reader in understanding the situation, we will suppose that he has arrived at Duluth or some other upper lake port to study the question.

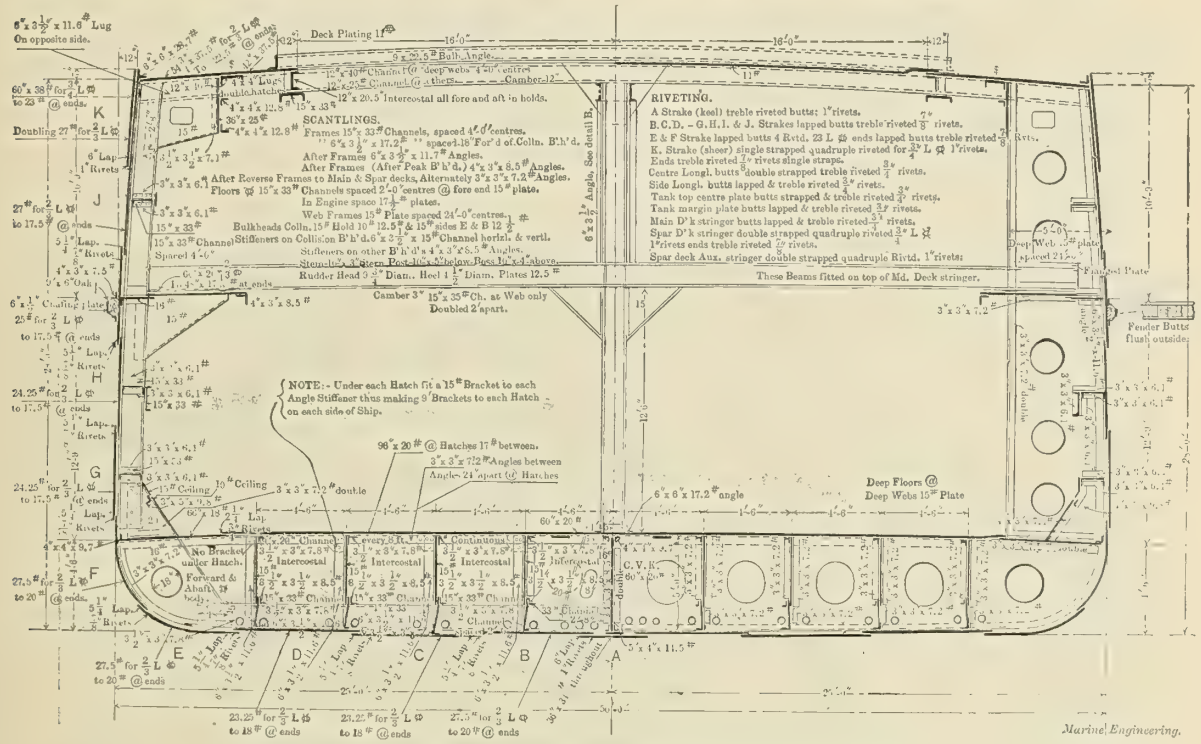
The grain as it arrives from the West and Northwest is received by the elevator for storage until it is again shipped. The elevator is a large, barn-like structure, impressing the observer rather with its bulk than with its beauty. Its purpose is to receive, discharge, and hold grain in storage. Structurally, the elevator consists of an assembly of parts and suitable machinery for elevating and discharging grain, scales for weighing it, and bins or tanks for storing it. The grain as it is shoveled from the car is caught in the "lofter" pit and carried up by some conveyance system to the weighing hopper, where, after a certain weight is received and recorded, it is carried by gravity or by some system of conveyance to storage bins; in some cases passing through cleaning and drying operations on the way. It may, however, be transferred to vessels or to other cars through the discharging spouts. The accompanying sections of modern grain elevators will serve to aid the reader in understanding the construction.

We will now suppose that a *freighter* of the largest size—say 440 feet long—has arrived at the elevator for a load of grain. The hatch covers are taken off, and, the preliminary work of adjusting the discharging spouts being over, the grain is allowed to flow into the hold of the vessel through several hatches at once. As the season of navigation is limited by ice to about seven months in the year, despatch in port is of the utmost importance and bulk cargoes are all spouted in—coal and ore from elevated dock structures—and taken out by machinery at the dock, no hoisting machinery being carried by the boat itself.

The cargo has now been received, and, the work of trimming being over, the vessel starts on her way eastward to Buffalo, the great eastern terminus of the lakes'

traffic. The movement of grain and ore is entirely from upper to lower lake ports, and of coal in the opposite direction. As, however, there are a great number of wooden vessels still in the service which require up cargoes, and as coal cannot be unloaded rapidly and the freight rate being generally quite low, the big steel vessels, especially those belonging to the great steel and transportation companies, go back light or use water ballast when any is required.

The majority of modern lake steamers are designed to carry only grain, ore, and coal in bulk, and so no deck is needed or is laid on the main-deck beams in the cargo hold. Due to the fact that all bulk cargoes are spouted in, the hatches are many in number and very wide. As the connecting channels between the lakes are comparatively shallow, the vessels are built with flat-plate keel and very little rise to the floor and very full model, so as to get the maximum carrying capacity on the limited draft available. On arriving at an elevator or loading dock in a light condition, these vessels, therefore, float very high out of the water, and, to obtain the necessary slope for the spouts and elevator "legs," the hatches are made as wide as possible transversely, leaving the stringer on each side to take care of the strength (the hatches actually being from 24 to 30 feet wide in vessels of 48 and 50 feet beam). Sometimes the water bottom is filled, thus depressing the deck; but this, in general, is not allowable, since it is evident that a cargo can be spouted in much faster than the water can be pumped out.



MIDSHIP SECTION OF A TYPICAL FREIGHT STEAMER OF THE GREAT LAKES.

SCANTLINGS.

Frames 15-inch by 33-pound channels, spaced 4 feet 0 inches centers.

Frames 6-inch by 3 1-2-inch by 17.2-pound channels, spaced 18 inches forward of collision bulkhead.

After frames 6-inch by 3 1-2-inch by 11.7-pound angles.

After frames (after peak bulkhead) 4-inch by 3-inch by 8.5-pound angles.

After reverse frames to main and spar decks alternately 3-inch by 3-inch by 7.2-pound angles.

Floors, midship section, 15-inch by 33-pound channels, spaced 2-foot 0 inch centers at fore end, 15-pound plate.

In engine space 17 1-2-pound plates.

Web frames, 15-pound plate, spaced 24 feet 0 inches centers.

Bulkheads, collision, 15 pounds; hold, 10 pounds, 12.5 pounds, and 15 pounds; sides, E and B, 12 1-2 pounds.

Stiffeners on collision bulkhead, 6 inches by 3 1-2 inches. 15-pound channel, horizontal and vertical.

Stiffeners of other bulkheads, 4-inch by 3-inch by 8.5-pound angles.

Stem, 10 inches by 3 inches; stern post, 10 inches by 5 inches below boss, 10 inches by 4 inches above.

Rudder head, 9 3-4 inches diameter; heel, 4 1-2 inches diameter; plates, 12.5 pounds.

RIVETING.

A strake (keel) treble-riveted butts, 1-inch rivets.

B, C, D, G, H, I, and J strakes lapped butts, treble riveted, 7-8-inch rivets.

E and F strakes lapped butts, 4 riveted, 2-3 L, midship section; ends lapped butts, treble riveted, 7-8-inch rivets.

K strake (sheer) single-strapped, quadruple riveted for 3-4 inch L, midship section, 1-inch rivets.

Ends treble riveted, 7-8-inch rivets, single straps.

Center, longitudinal butts, double strapped, treble riveted, 3-4-inch rivets.

Side, longitudinal butts, lapped and treble riveted, 3-4-inch rivets.

Tank top-center plate butts strapped and treble riveted, 3-4-inch rivets.

Tank margin--plate butts lapped and treble riveted, 3-4-inch rivets.

Main deck stringer butts lapped and treble riveted, 3-4-inch rivets.

Spar deck stringer double strapped and quadruple riveted, 3-4 L, midship section, 1-inch rivets. Ends treble riveted, 7-8-inch rivets.

Spar deck auxiliary stringer double strapped and quadruple riveted, 1-inch rivets.

Coal and ore docks have their spouts and unloading apparatus uniformly spaced twelve feet centers, so we find all hatches on lake crafts spaced 24 feet centers and usually 8 feet long; however, there are a few boats now with hatches spaced 12 feet centers. The stanchions supporting the spar deck between the hatches are of channel section and are as few in number as is consistent with the weight to be supported and general strength considered.

The frame spacing in the latest construction in the larger type of lake vessel is 4 feet above the tank and 2 feet for the floors below—both frames and floors being of channel section. In most of the large boats a deep web frame, with corresponding deep-plate floor, is fitted clearly, and in this connection the reader should note the universal use of shaped sections whenever possible.

To avoid the interruption of the loading spouts, as well as to avoid the inconvenience of a shaft alley in the cargo hold, the machinery is placed aft—the engine going as near the stern post as the shape of the vessel will permit, and the boilers and bunkers are placed next forward, either on the tank top or on a special deck with cargo space beneath. The longest straight run in open water is across Lake Superior, a distance of about 400 miles, and the total voyage between, say, Duluth and Buffalo, being under 1,000 miles; and as coal for fuel is available at various points in the connecting channels, the coal supply is not great. Recently, however, atten-

UNLOADING GRAIN INTO ELEVATORS.
[Copyright, 1900, Detroit Photographic Co.]

every 24 feet and arranged to come between the hatches. For convenience in handling various cargoes the large hold space is usually divided into three or four compartments by screen bulkheads.

The danger of grounding in the narrow and, in many places, rocky channels connecting the lakes is very considerable, and double bottoms are a necessity and are carried straight across on top to protect the bilges to the upper turn. To get sufficient water ballast, as well as to raise up a heavy cargo, such as wheat or ore, and make the ship easier in a seaway, the tank is made deep—from 5 to 6 feet at the center line. In carrying out the design, great care must be exercised in securing proper strength in the water bottom to sustain the load and also to support the bottom in the event of grounding. The midship section shows this construction very

tion has been called to the high price of fuel at upper lake ports, so that large bunkers are arranged.

The machinery of the boat is modern in all particulars and consists of two or three Scotch boilers supplying steam at 150 to 180 pounds pressure to a triple-expansion engine and the necessary auxiliary machinery, including ballast pumps of large capacity. Some of the vessels of this type and size have been fitted with water-tube boilers and quadruple-expansion engines; but this equipment is not common. The horse power of the engine is comparatively low when the size of the boat is considered, but is ample to propel the vessel on a draft of 18 feet at from 10 to 12 miles per hour, a speed found to be the most economical for the lake freighter.

The accommodations for the officers and crew and occasional passengers are at the extreme forward and

after ends of the ship, thus leaving the working deck free from all obstruction. However, some boats have a small deck house amidships. In the forward part of the ship are usually provided the owner's suite of rooms and quarters for the mates, wheelsmen, and watchmen; while above in the Texas, adjoining the pilot house, the captain has his apartments and chart room. The after deck house, forming part of the boiler house, contains the galley, pantry, dining room, crew's mess, and quarters for the engineers, oilers, and stewards, with occasionally a spare room or two. The deck hands and firemen are usually quartered below, just aft of the engine hatch, on a partial deck or on the level of the handling deck.

During these observations the vessel has approached the Sault Ste. Marie ship canal, commonly known as the "Soo," and after the necessary operations involved in passing the locks are performed, the ship steams on her way eastward. We will here present a few bits of information concerning the Soo which will give the reader a good idea of the extent of the lake traffic. At the close of this season it is reported that about 27,000,000 tons of ore passed through the canal, about 3,000,000 tons of wheat, and about 6,000,000 tons of general freight. Without exaggeration there is probably no other highway through which a greater amount of commerce passes.

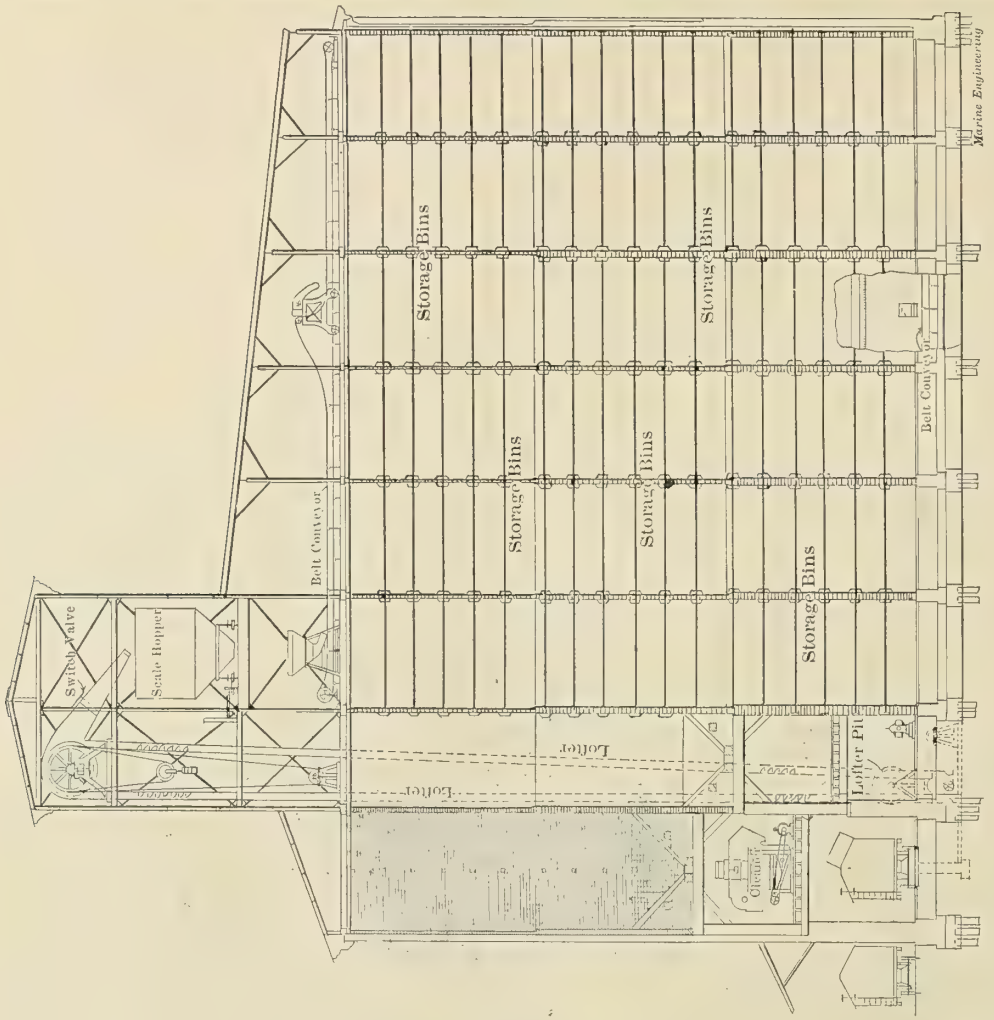
All along the route the observer sees evidences of the

GREAT NORTHERN ELEVATOR AND SHIPPING AT BUFFALO, N. Y.
[Copyright, 1900, Detroit Photographic Co.]

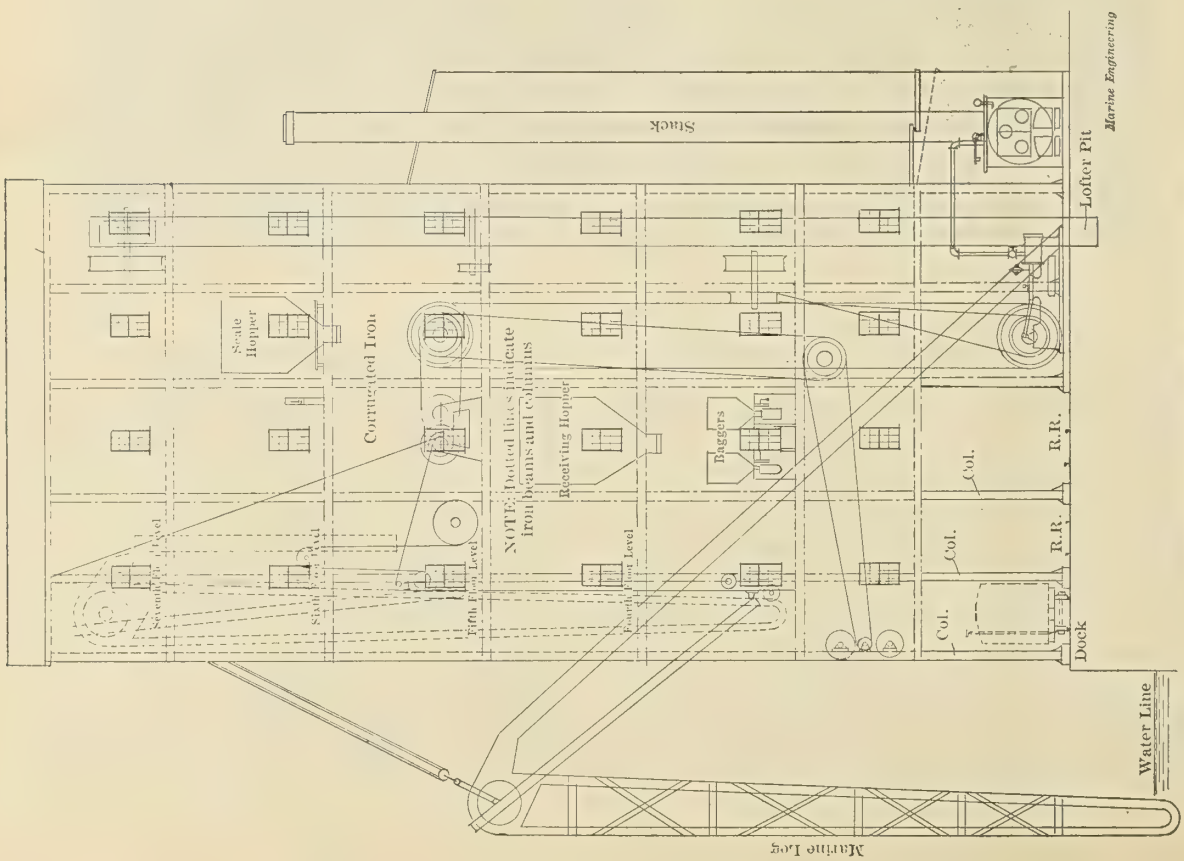
The joiner work in the cabins presents a finished and pleasing appearance to the eye, and the oak and other hard woods usually employed suggest an idea of comfort and endurance. Electric lighting is used throughout the vessel and a searchlight is provided for making docks, etc. All living quarters are heated by steam, and in general fittings and equipment the boat is not at all inferior to any coast vessel engaged in similar trade. We would here remark that, due to a misunderstanding of the situation, the lake freighter is generally considered and specified on the coast, in respect to size and equipment, as a barge, and so great surprise is expressed by salt-water sailors on inspecting for the first time one of these barges 450 to 480 feet long. Taking everything into consideration, the vessel presents from the outside a not ungraceful appearance with her slender pole masts and smokestack raked appreciably aft.

vastness of the steel and iron industry of our country in the numerous freighters of the largest size. Many of these have silver-colored stacks, which are the distinctive features of the Pittsburg Steamship Company's fleet, controlled by the United States Steel Corporation. Now and then a whaleback steamer passes by and attracts the observer's attention by its peculiar appearance. These boats were quite popular a few years ago, but, due to several reasons, they are no longer built. Here an oldtime schooner comes in sight, a relic of the enormous sailing fleet that once covered these waters; there a great log raft is seen with several tugs steering it slowly along, an evidence of the vast lumber traffic of the Great Lakes.

We have now arrived at Buffalo and the ship makes her way to some particular elevator, according to her contract, and prepares to discharge her cargo. The



BRICK ELEVATOR, 1,000,000 BUSHELS CAPACITY.



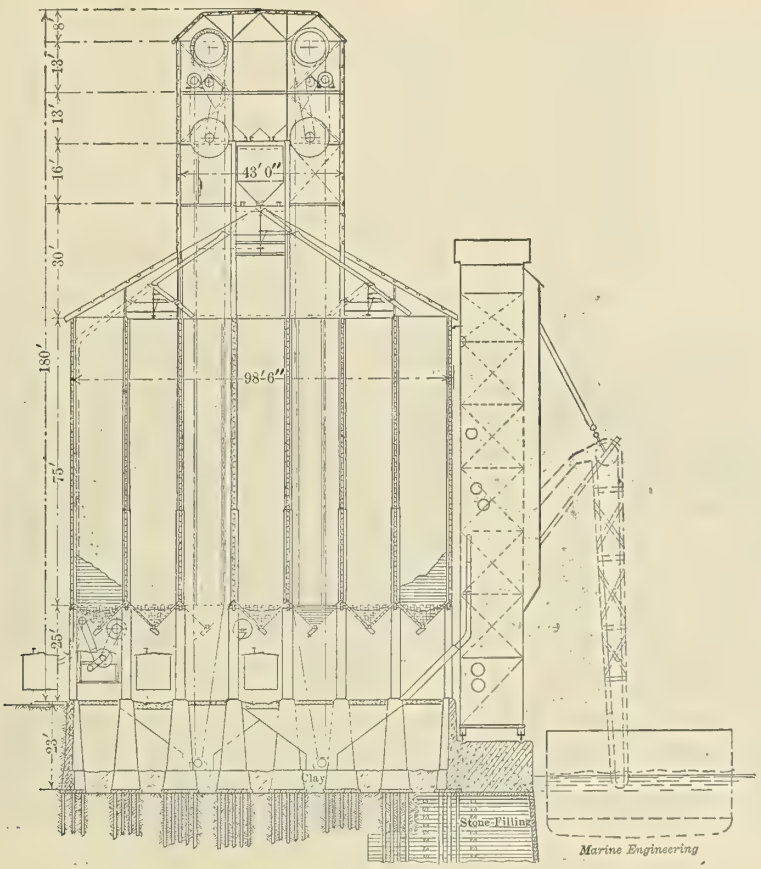
WOODEN ELEVATOR, 750,000 BUSHELS CAPACITY.

hatch covers are lifted off and one or more marine elevators, or "legs," are lowered into the hold. The marine leg is a framework or steel structure supported at the elevator end, so as to allow the other end, which rests in the hold of the vessel, to adjust itself to the varying levels of the grain. The leg carries a system of buckets or pockets, which are fitted to an endless belt actuated by power received through rope gearing from the power plant of the elevator. The buckets, as they travel up and down the leg, catch up the grain and discharge it at the top of the leg into a chute leading down to the lofter pit, where it is caught up as before and carried to the weighing hopper. These legs take out grain with great rapidity, each one being able to elevate from 15,000 to 18,000 bushels per hour. When the level of the grain has been lowered to within a short distance of the tank top, steam shovels are fitted and worked through rope purchases from the elevator, so as to collect the grain about the leg. Down in the hold, in the dust-laden atmosphere—so dense that one can hardly see—gangs of men with ears, eyes, and nose covered with some sort of protection, and armed with large scoop shovels, feed the steam shovels. This work is very trying, and none but experienced and powerful men can stand it long, but a strong inducement is offered in the wages, which amount to about 40 cents per hour.

The vessel now leaves the elevator and probably looks around for an up cargo of coal or returns to Duluth for another cargo of grain.

The grain is held in storage or transferred to cars or to canal boats. These latter, having received their cargo, commence their slow journey down the canal, and on arriving at New York discharge at certain points for domestic use or for exportation abroad.

Before closing we will present a few facts concerning Buffalo as a great commercial center, and the economics of the grain trade. During the last season the shipping tonnage arriving at Buffalo amounted to 5,300,000 tons, and the amount of grain received reached 130,000,000 bushels. There are 24 large elevators in Buffalo, with a total capacity of about 20,000,000 bushels.



PROPOSED STEEL ELEVATOR, 1,000,000 BUSHELS CAPACITY.

Besides being a great lake port, Buffalo is one of the largest railroad centers, having trunk line connections to all parts of the country. The grain (let us take wheat for example), as it arrives at Duluth or other upper lake port, is probably worth from 50 to 70 cents per bushel. The elevation of the grain amounts to from one-quarter of a cent to one cent per bushel. The capital invested in the elevator varies from 18 cents to 30 cents per bushel capacity, respectively for wood and steel elevators, while the capital invested in the grain carrier varies from, say, $1 to $1.25 per bushel capacity of the vessel. The trimming of the grain in the vessel's hold is paid for by the boat and varies at different ports—at Duluth it amounts to 50 cents per 1,000 bush-

els, while at Chicago it is 75 cents per 1,000 bushels. The cost of transportation varies from 1 1-2 to 3 cents per bushel, depending on the season and market. In the old days of sailing vessels very high rates were sometimes paid, and in 1872 we find a rate of 25 cents per bushel from Chicago to Buffalo. All unloading is done by contract, to be paid by the boat—that is, the cost is deducted from the freight bill. The question of "overage" and "shortage" is settled by the terms of the contract, which always guarantee delivery at the receiving elevator of a certain definite amount. Should a shortage happen the boat must make good the deficit at cost price of the grain, while should an overage result the boat is paid for the amount at regular price by the elevator.

CANAL BOATS WHICH HAVE BEEN TOWED THROUGH THE ERIE CANAL AND HUDSON RIVER, IN THE EAST RIVER, NEW YORK CITY.

New York Fireboat.

After a lively competition among shipbuilders the City of New York placed with the New York Shipbuilding Company, Camden, N. J., an order for a fireboat for the Fire Department of the Boroughs of Manhattan and the Bronx.

The vessel is 117 feet over all, and other dimensions will be found in a table at the end of this article. The specifications call for a vessel complete in every respect, excepting the fire hose and nozzles and the cabin equipment. All steel used is to be open-hearth steel. and the material supplied is to be tested in accordance with the rules laid down by the American Shipmasters' Association. The bar keel is of extra-rolled bar iron 6 by 1 1-2 inches, worked in lengths of about 25 feet, with 18-inch scarfs. The stem is of 6 by 1 3-4-inch bar iron, rabbetted to receive the plating to about 4 feet below the water line.

The stern post is of cast steel, 6 by 3 1-2 inches, and the rudder post is 6 by 3 inches. The rudder is of forged iron, 5 inches at head, 2 1-2 inches at heel, and made in one piece with two arms. The weight is supported by three pintles and the plating is of 10 pounds, the space between being filled with white pine.

The tiller, with proper chains, is fitted upon the rudder stock below the level of the main deck.

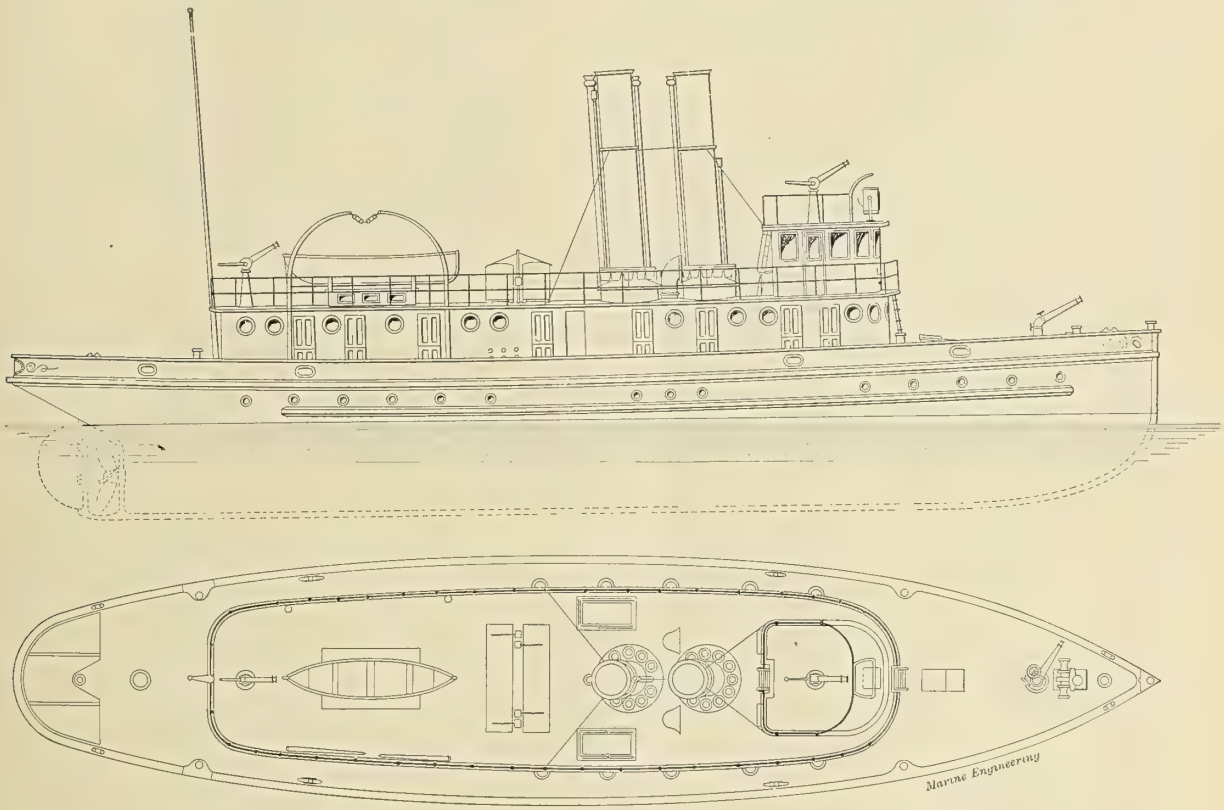
The frames are 3 by 2 1-2 inch by 5 1-2-pound steel angles, spaced 20 inches, and scarf pieces 2 feet 6 inches in length are to be fitted across the line of keel back of floor plates on every frame. The reverse frames are 2 1-2 by 2 1-2 inch by 3 1-4-pound angles, extending alternately to the under side of the deck-beam bracket and to the upper stringer angle. The floors are 15 inches deep, of 12 1-2 pound plating, and under the engine extend to the engine seating, and under the boilers two floors to be extended to form saddles for same. A 9 by 4.4-inch by 23.4-pound I-beam forms the center keelson and is fastened to reverse frames with clips. The bilge keelson and hold stringer run as far forward and aft as practicable, and each is formed by a pair of 3 by 2 1-2 inch by 5 1-2-pound angles riveted to reverse frames and clips. At the forward end they are secured by gusset plates to the collision bulkhead. Over the engine and boiler foundation 12 1-2-pound intercostal plates, fastened by 3 by 2 1-2 inch by 4 1-2-pound clips, are worked between the floors.

The plating is worked in six in-and-out strakes. All

laps are single riveted, except the garboard which is double riveted. The butt straps are double riveted. The water-line strake at the bow is double plated for a distance of 10 feet from the stem. The garboard strake and sheer strakes for three-fifths length are 16 pounds; two-fifths of the sheer strake and three-fifths the length of the side and bilge strake are of 14 pounds; the remaining lengths of the side and bilge strakes are of 12-pound plates. The bulwarks are of 10-pound plating, supported by stanchions 1 1-2 inches in diameter. Three cast-iron scuppers and three freeing ports to drain the waterways are located on each side.

The deck beams are of bulb angles, 6 by 3 inches by 12.3 pounds, placed on alternate frames and to be stiffened. The tops of the peak tanks are covered with 3-16-inch plate. A tank is constructed by extending the floor plate on the floor 10 feet forward of second bulkhead and plating back to bulkhead with 3-16 inch plate. All tanks are fitted with manholes and proper filling and draining valves, etc., and piping connected to a manifold.

The guards are to be two in number, the upper one of white oak 7 1-2 inches thick by 9 inches wide, to be carried from the stem along the gunwale round the stern; the lower guard is of white oak 6 inches thick by 5 1-2 inches wide. The upper one is stiffened by double 3 by 3 inch by 6.1-pound angles, and the lower by 3 by 2 1-2 inch by 5 1-2-pound angles. Chafing



FIREBOAT DESIGNED BY H. DE B. PARSONS FOR THE CITY OF NEW YORK.

fitted with welded knees. The crown is 6 inches in 24 feet. Where half beams are placed they are to be made of double angles, 3 by 2 1-2 inches by 5 1-2 pounds. The lower-deck beams are 3 by 2 1-2 inch by 4 1-2-pound angles. Hold stanchions are of 2-inch forged iron. The main-deck stringer plates, which are 21 inches by 11 pounds, tapering to 18 inches by 10 1-2 pounds, are to be worked on the main deck. The gunwale angles are 3 by 2 1-2 inches by 5 1-2 pounds, and the waterway angles 3 by 2 1-2 inches by 4 1-2 pounds. Four transverse bulkheads are fitted, the forward one water-tight and forming the forepeak, of 12 1-2-pound plating, the second of 7 1-2-pound plating, the third extending between the coal bunkers, of 7 1-2-pound plating, the fourth to be of 12 1-2-pound plating, forming the afterpeak tank. All bulkheads are properly stiffened. The wing coal bunkers are of 10-pound plating properly bands are to be worked on the outer edge of each guard. Hawse pipes are to be galvanized iron. The mooring bitts are to be placed four on each side, and two score chocks are to be located at the bow and two at the stern, made of composition. A "nigger-head" of galvanized cast iron, with composition cap, is to be placed at the stem, and there are four deck cleats of cast iron galvanized, two on each side. A galvanized hand windlass is securely bolted to forward deck plate. There are four coal scuttles and one ash scuttle on each side.

The main-deck planking is of selected white pine, 3 by 3 inches, laid parallel to gunwale.

The deck house is to be built, as shown in the plans, 7 feet high in the clear and plated with 8 1-2-pound steel, the framing to be of 2 1-2 by 2 1-2-inch by 4.2-pound angles. The house is to be attached to inside

stringer by an angle 3 by 2 1-2 inches. The beams of the deck house and pilot house are of 3 by 2 1-2-inch by 4 1-2-pound angles. The house decks will be of white pine, 1 1-8 inches thick by 3 inches wide, covered with 10-ounce cotton canvas.

The rail is of clear, selected white oak, 10 inches wide by 3 inches thick, with rounded edges and bolted to angle bar. The pilot house is to be ceiled with sycamore. The forecastle will be fitted with ten folding galvanized iron berths; the port lights, numbering twenty-eight in all, are 10 inches in diameter in the clear.

There will be a steam steeering gear with small wheel, and also a hand gear with a larger wheel, so arranged that the hand gear can be easily thrown in when necessary. One 18-foot metallic lifeboat is to be supplied also anchor davit, boat davits, 300-pound folding anchor, necessary anchor chains, etc., to meet the government requirements. All living quarters are steam heated.

MACHINERY.

The propelling machinery consists of an inverted compound engine, with cylinders 17 and 34 inches in diameter by 22-inch stroke. Steam is supplied by two Scotch boilers of dimensions given in the table. The three pumps for fire service are supplied by the Fire Department, and are of the vertical, simple, duplex, flywheel, direct-acting type, two having steam cylinders 17 inches in diameter, water cylinders 10 inches in diameter, by a common stroke of 11 inches; the cylinders for the other pump being 9, 5 1-2 by 8 inches.

The high-pressure cylinder is fitted with a piston valve and the low-pressure with a double-ported balanced slide valve. The pistons are of cast iron, of the dished pattern, with cast-iron rings and springs. The engine is supported at the back by cast-iron columns fitted with crosshead slides, and the front columns, two in number, are of wrought iron. The bed plate is of box section cast iron and fitted with four main bearings. The condenser is of cast iron and forms part of the framing, and the tubes are of solid-drawn brass, tinned, 5-8 inch in diameter and No. 19 B.W.G. There is a steam reversing engine, which may be operated by hand. The air and circulating pump is of the Blake pattern, with one double-acting steam cylinder 16 inches in diameter, with two single-acting air cyclinders 12 inches in diameter with a stroke of 8 inches. The circulating pump, of the centrifugal type, has a capacity of 1,000 gallons per minute and will be driven by a 5 1-2 by 5-inch engine. The piston rod is of mild steel; the crosshead is of the same material with journals forged on and bolted to cast-iron shoe. The connecting rods are of steel, and the crank pins are 7 3-4 inches in diameter and 8 1-2 inches long. The crank shaft is of the built-up type; the thrust shaft is 7 1-4 inches in diameter and is to be fitted with six cast-iron horseshoe collars; the propeller shaft is to be of hammered iron, 7 inches in diameter, with a single composition casting shrunk on. The stern tube is of cast iron bushed with composition and fitted with lignum vitæ staves. The main steam piping is of seamless-drawn copper, 7 inches in diameter, with 5-inch branches to boilers and 5-inch branches to each large fire pump; exhaust pipe to be of copper, 11 inches in diameter. The suction and boiler-feed piping is also to be of copper.

A duplex bilge pump, 6, 4 by 7 inches, is to be connected to the bilge and water tanks. The feed pump will also be a duplex pump of the Blake or other approved type, with cylinders 6, 4 by 7 inches. An electric generator of 9 K.W. capacity will supply light throughout the vessel at 110 volts. There will also be located on the top of the pilot house a searchlight with an 18-inch projector.

NAME.—

LAUNCHED.—

NATIONALITY—American.

OWNER—New York City.

TYPE—Fireboat.

BUILDER—New York Shipbuilding Co.

Length over all	117 ft.
" on water line	110 "
Beam, molded	24 " 3 1-2 ins.
Draft, load	9 " 6 "
Engines	one compound.
Cylinders diam.	17, 34 ins.
Stroke	22 "
Revolutions, estimated	150
H. P., guaranteed	500.
Valves	piston, H. P.; slide, L. P.
Piston rods, diam.	4 ins.
Crossheads, wearing surf	90 sq. ins.
Connecting rods, length	60 ins.
Crank pins	7 3-4 ins. diam. by 8 1-2 long.
Shafting, crank and thrust	7 1-4 ins.
Propeller shafting	7 ins.
Condenser, cooling surf	1,150 sq. ft.
Propeller	Solid cast iron, four blades.
Diameter	7 ft. 6 ins.
Pitch	9 " 9 "
Boilers	two Scotch.
Diameter	10 ft. 9 ins.
Length	11 " 9 5-8 "
Pressure	140 lbs.
Furnaces	two Morison in each.
Diam., inside	40 ins.
Thickness	7-16 in.
Tubes, in each boiler	170
" of charcoal iron, No. 12 B. W. G.,	diam. 3 ins.
Stay tubes, thickness	1-4 in.

Refrigeration.—To cool and purify the air in ships' holds, an apparatus has been patented in England by C. W. Bollman by which the air is drawn from and returned to the hold after being cooled. The air is passed through a casing, where it is subjected to a spray of cooled brine from perforated pipes. Circulation of air is secured by a fan.

Cost of Government-Built Ships.—In the hope of lessening the expenses of the government in the contest with the Newport News Shipbuilding and Dry-Dock Company to determine whether it is possible to build warships at the navy yard as cheaply and quickly as a similar vessel is built by a private firm, Rear Admiral Bowles has recommended to Secretary Moody that the men engaged on the battleship *Connecticut*, at the New York Navy Yard, be paid by piecework instead of by the day. In private yards a man works ten hours a day and is paid by piecework, while in the government yards he is paid by the day of eight hours.

THE POWERING AND PROPULSION OF SMALL BOATS.

BY W. F. DURAND.

In the present chapter it will be our purpose to give some general discussion of methods for determining the amount of power required for the propulsion of small boats at such speeds as may be desired. The shape and proportions of the boat enter largely into the result, and the problem itself is so complex that, at best, no absolute rules can be given. In all such cases judgment must enter in very largely, and the most that can be done is to give some orderly procedure whereby judgment may be used to the best advantage with the material at one's disposal. For the purpose of this discussion we shall use the usual Admiralty horse-power formula in the following form:

$$H = \frac{A\,B}{K} \qquad (1)$$

in which H is the horse power required; the values of A, corresponding to the weight of the boat when fully loaded, is taken from Table I; that of B, corresponding to the speed in miles per hour, from Table II; and that of K selected by judgment, as discussed below.

TABLE I.—VALUES OF A.

Weight, pounds.	A	Weight, pounds.	A
500	.368	12,000	3.05
1,000	.584	13,000	3.23
1,500	.765	14,000	3.39
2,000	.927	15,000	3.53
3,000	1.21	16,000	3.71
4,000	1.47	17,000	3.86
5,000	1.71	18,000	4.01
6,000	1.93	19,000	4.16
7,000	2.13	20,000	4.30
8,000	2.33	21,000	4.44
9,000	2.53	22,000	4.58
10,000	2.71	23,000	4.72
11,000	2.88	24,000	4.86

TABLE II.—VALUES OF B.

Speed, miles.	B	Speed, miles.	B
5.0	125	11.5	1,521
5.5	166	12.0	1,728
6.0	216	12.5	1,953
6.5	275	13.0	2,197
7.0	343	13.5	2,460
7.5	422	14.0	2,744
8.0	512	14.5	3,049
8.5	614	15.0	3,375
9.0	729	15.5	3,724
9.5	857	16.0	4,096
10.0	1,000	16.5	4,492
10.5	1,158	17.0	4,913
11.0	1,331	17.5	5,359
		18.0	5,832

It may be noted that the values of A are found by taking the cube root of the square of the weight, reduced to tons of 2,240 pounds. Thus, for a total weight of 8,000 pounds we have 8,000 ÷ 2,240 pounds, or 3.57. The square of this is 12.74, and the cube root of this is 2.33, as noted in the table opposite 8,000. Other corresponding values may thus be found if desired. The values of B are also the cubes of the speed in miles per hour, and other or intermediate values may be found as desired. The values of K must be taken by judgment, guided by the following general indications: The value to be selected will depend chiefly on the three items, length, speed, and form. Other things being equal, it will be greater as the length is greater, less as the speed is greater, and greater as the form is finer; or, somewhat differently, its value will be less as the proportion of speed to length is greater, and as the form is fuller or proportion of beam to length is greater; and its value will be greater under opposite conditions. A short, wide, full form, at relatively high speed, would, therefore, call for the smallest values of K, or for relatively the largest values of H, as common-sense would tell us; while a relatively long, slender, fine form, at moderate speed, will call for relatively less power, and hence for a large value of K. As to what we mean by a speed relatively high or low in comparison with the length, there is no very definite rule. In general it will not be far wrong to consider any speed in miles per hour which is numerically less than the square root of the length of the boat in feet as relatively low, and *vice versa*. Thus, for a boat 25 feet long, 5 miles or anything less is relatively a low speed, while 6 and upward are relatively high; while again, for a fifty-foot boat, speeds of 7 or 8 miles are low, and those of 9 or 10 and upward relatively high.

The actual values for small craft such as we are now considering, and at ordinary full speeds, are usually from 150 to 225 or upward. For large-sized ships, at moderate speeds, the values are much larger, rising to 400 and over in special cases. It should be noted that these values refer to speed expressed in miles per hour, as is common in discussing small boats. If the speed is expressed in knots, the values for small boats would become reduced to 100 to 150, rising to 200 and upward to 300 or more for large ships in special cases.

In illustration of the above suggestions we may assume a boat of 50 feet length at 7 miles per hour. Here the speed is moderate and a value of 225 would not be unreasonable. As this speed is exceeded, however, the corresponding value of K should decrease in order to provide for relatively greater power, and with a form of moderate fineness may be expected to come down to somewhere about 150 for a speed about twice the square root of the length in feet, or at 14 to 15 miles. For a 25-foot boat the corresponding speeds would be 5 and 10 miles respectively.

The influence of form on this value, as referred to above, must also be borne in mind. The two chief aspects are the relation of length to beam and the degree of longitudinal fullness. The narrower the boat for the length, and the finer the form longitudinally, the greater the value of K, and *vice versa*.

We must next note the relation between the dimensions of the boat and the probable displacement or total weight, as referred to in Table I. The actual volume of water displaced will be a certain fraction of the box formed on the chief dimensions, length, beam, and draft. This fraction is known as the block coefficient. Its value is usually found between .40 and .60 or .65 for boats of the type we are now considering. The lower value .4 would correspond to a fine-lined racing launch, while the larger values would correspond to fuller boats, in which comfort and capacity are of more importance than speed. Values of .55 to .60 are quite common in construction of this character. Then, remembering that sea water weighs 64 pounds per cubic foot and fresh water 62.5 pounds, the total displacement

in pounds is readily found from the estimate of the volume thus made.

A few illustrative examples will serve to show the method of applying these principles to the determination of displacement and power.

Suppose length 25 feet, beam 5 feet, and mean draft 2 feet in salt water, with a fairly full form. A speed of about 8 miles per hour is desired. What power should be provided?

Taking the block coefficient at .65, we have for the volume of the under-water body $25 \times 5 \times 2 \times .65 = 162.5$. Multiply this by 64 and we have for the weight $64 \times 162.5 = 10{,}400$ pounds. Taking the value of K at 180 for this case, and the corresponding values of A and B from the tables, we have:

$H = 2.78 \times 512 \div 180 = 7.9$, or, say, 8 horse power.

Again, take length = 20 feet, beam 5 feet, mean draft 18 inches in fresh water, speed 6.5 miles per hour. Take the block coefficient the same as above, and we have for the probable displacement, $20 \times 5 \times 1.5 \times .65 = 97.5$ cubic feet. Multiply this by 62.5 and we have 6,094 pounds. Taking the same value of K as above gives:

$H = 1.95 \times 275 \div 180 = 2.98$, or 3 horse power.

Again, take length = 40 feet, beam 5 feet 6 inches, mean draft 2 feet 6 inches in salt water, speed 12 miles per hour. Assuming that this is a boat built for speed and with a finer form than those above, we may take the block coefficient at .45. Then for the displacement we have: $40 \times 5.5 \times 2.5 \times .45 = 247.5$ cubic feet. This multiplied by 64 gives 15,840 pounds.

For this case, taking $K = 210$, we have:

$H = 3.68 \times 1{,}728 \div 210 = 30.3$, or about 30 H. P.

While there will be need for the exercise of much judgment based on experience, the foregoing suggestions will at least serve to give a reasonably close approximation to the power which should be supplied in any case likely to arise in ordinary experience.

DESIGN OF PROPELLER.

This problem is, if anything, still more complicated than that of determining the necessary power, as discussed above. We shall give, however, a relatively simple procedure from which satisfactory results may be obtained in all ordinary cases.

Let d = diameter in feet.
p = pitch in feet.
N = revolutions per minute.
H = horse power.
u = speed in nautical miles per hour.
K = coefficient taken from Table III, in accordance with the pitch ratio or value of $p \div d$.

Then we have the following equations:

$$d = K \sqrt[5]{\frac{H}{u^3}}$$

$$pN = 117\, u.$$

TABLE III.

Pitch Ratio.	K
1.0	12.8
1.1	13.6
1.2	14.3
1.3	15.1
1.4	15.8
1.5	16.6
1.6	17.3
1.7	18.0
1.8	18.8

These values are intended primarily for a four-bladed wheel, with generally oval blades, the total blade area being from .36 to .45 of the disk area or of the area of the circle $\pi d^2 \div 4$. Values of d less than those given by the equation should not be adopted unless necessary. If, however, the available room will not admit the diameter thus found, it may, with some sacrifice of efficiency, be reduced, and with it the other features of the wheel proportionately, to an amount which should not exceed 15 to 20 per cent., while correspondingly it may be expected that the slip will increase and with it the value of pN by about 10 per cent., the decrease of d and increase of pN keeping relative step, the latter being somewhere about one-half the former.

Furthermore, to use these results for *two* or *three* blades instead of *four*, the following relations may be observed:

For two blades, other things being equal, the diameter found as above should be increased by about 20 per cent., or one-fifth.

For three blades, other things being equal, the diameter found as above should be increased by about 10 per cent., or one-tenth.

The use of these equations may best be illustrated by a series of examples.

Taking first example 1 above, we have $H = 8$ and $u = 8$. We will try first a pitch ratio of 1.2. Then we have:

$$d = 14.3 \sqrt[5]{\frac{8}{512}} = 1.8 \text{ feet} = 21.6 \text{ inches.}$$

$p = 1.8 \times 1.2 = 2.16$ feet $= 25.9$ inches.
$pN = 936$.
$N = 936 \div 2.16 = 433$.

It is quite likely that this value of N may be considered too high. In such case we must choose a larger pitch ratio. Taking, therefore, a value 1.3 for further trial, we find similarly:

$d = 1.89$ feet $= 22.7$ inches.
$p = 2.46$ feet $= 29.5$ inches.
$N = 380$.

Suppose in this case, however, a two-bladed wheel is desired. Then the diameter should be increased by about one-fifth, or to about 2.27 feet. We should then have:

$d = 2.27$ feet $= 27.2$ inches.
$p = 2.95$ feet $= 35.4$ inches.
$N = 317$.

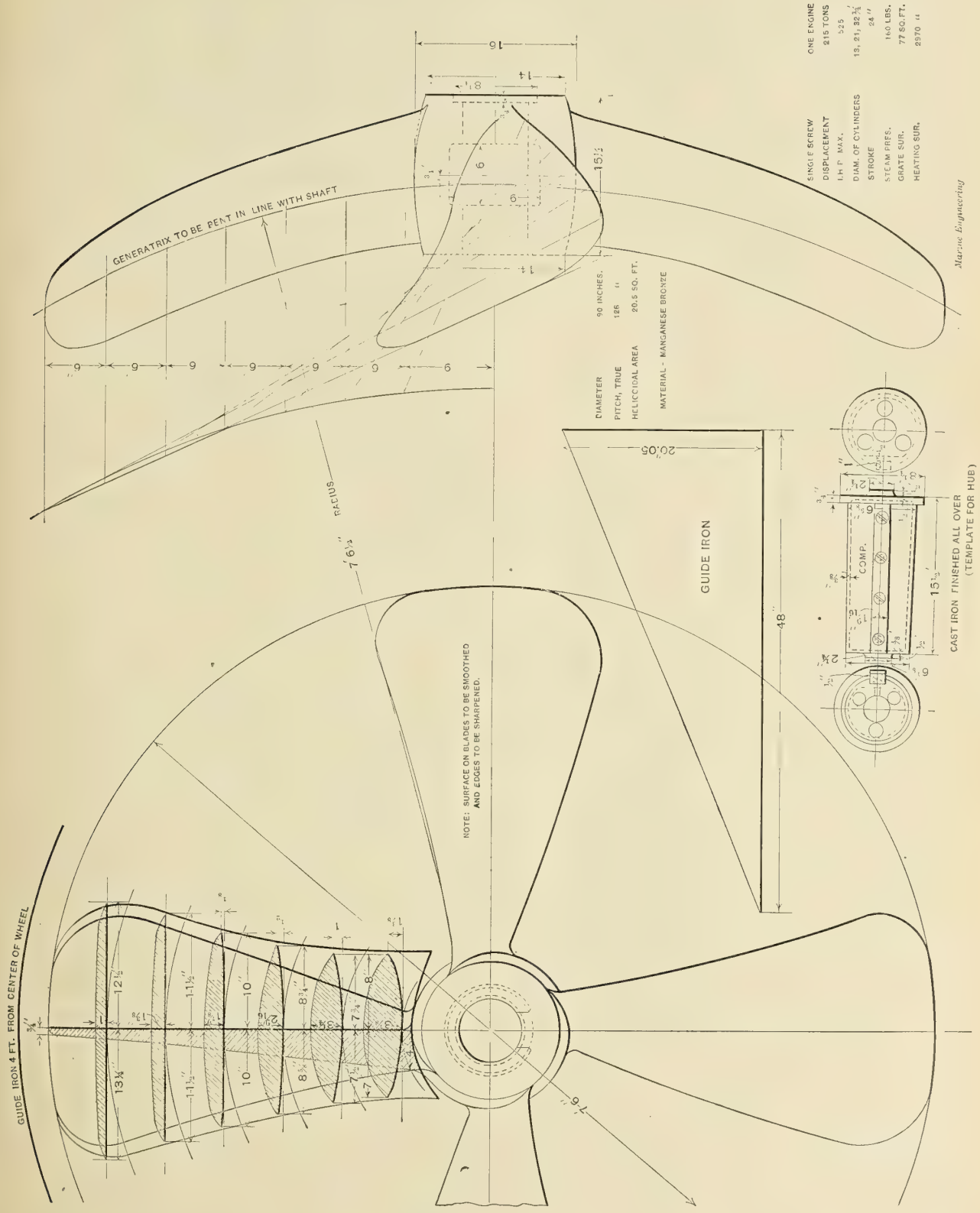
In such case it is quite likely that the lower pitch ratio 1.2 would be more appropriate, and taking our results for that ratio and enlarging d to correspond to a two-bladed wheel, we find as follows:

$d = 2.16$ feet $= 25.9$ inches.
$p = 2.59$ feet $= 31.1$ inches.
$N = 361$.

Take next the second boat assumed for illustration at an earlier point, and we have $H = 3$, $u = 6.5$. Taking on trial pitch ratio = 1.3, we find as follows:

$d = 1.58$ feet $= 19$ inches.
$p = 2.05$ feet $= 24.6$ inches.
$pN = 760$.
$N = 371$.

If, on the other hand, this is to be a two-bladed wheel, the values will result as follows:



FOUR-BLADED PROPELLER.

$d = 2.05$ feet $= 24.6$ inches.
$p = 2.66$ feet $= 32$ inches.
$N = 286$.

We may do something better with a pitch ratio 1.2, for which we find values as follows:

$d = 1.79$ feet $= 21.5$ inches.
$p = 2.15$ feet $= 25.8$ inches.
$N = 354$.

Suppose in this last case that it were desired to reduce the diameter somewhat, even at the expense of some propulsive efficiency. Let us then reduce the diameter to, say, 1.6 feet $= 19.2$ inches. This is a reduction of about 11 per cent., and would probably cause an increase of about 6 per cent. in the value of pN. We should then have results as follows:

$d = 1.6$ feet $= 19.2$ inches.
$p = 1.92$ feet $= 23$ inches.
$pN = 805$.
$N = 419$.

As a final example let us take the third boat assumed above. We have in this case $H = 30$, $u = 12$. Taking again as a first trial pitch ratio $= 1.5$, we find:

$d = 2.19$ feet $= 26.3$ inches.
$p = 3.29$ feet $= 39.5$ inches.
$pN = 1404$.
$N = 427$.

If this value of N is considered too high, as might be the case with engines of this power, we must take a still larger value of the pitch ratio. Thus for 1.6 we find as follows:

$d = 2.28$ feet $= 27.4$ inches.
$p = 3.65$ feet $= 43.8$ inches.
$N = 385$.

While for a value 1.7 we should find as follows:

$d = 2.37 = 28.4$ inches.
$p = 4.03 = 48.4$ inches.
$N = 348$.

These might be further modified for 2 or 3 blades, or for decrease of diameter with increase of slip and greater values of pN, and other examples might be given; but the preceding should be sufficient to illustrate the method as applied to the types of cases most commonly met with in connection with boats of the character under present discussion.

FORMS OF PROPELLERS.

In the preceding section relating to the design of propellers we have referred to the form simply as determined by the number of blades and by the generally oval form of the blade contour. In the present section we wish to give somewhat more detailed reference to the question of form and to the propeller as an engineering structure.

A screw propeller may be defined as a series of blades having on the rear or driving face an approximately helicoidal surface, such blades being secured to a common hub through which they are joined together and receive their motion of revolution from the shaft on which the hub is mounted. It follows that the blades in effect constitute the propeller, and upon their form and dimensions depends the entire performance of the propeller as an instrument of propulsion.

For our present purposes we may define a helicoidal surface as one generated by a straight line revolving about an axis at right angles to it, and moving its point of contact along the axis at the same time. It is in effect the same kind of surface as that which is found on the sides or flanks of a square-screw thread, as on the lead screw of a lathe. The face of a propeller blade is thus similar to a small fraction of a turn of a very deep screw thread, and this, with the thickness laid on, constitutes the blade itself. The propeller is then made up of two, three, or four such blades. The propeller blade, thus defined, is the simplest possible in form, producing the so-called true screw or uniform-pitch propeller with blades at right angles to the axis. The blades may, however, be bent or curved back somewhat, by a suitable modification of the generating line, and various modifications may be introduced into the form of the blade, all without changing its character as a true screw or propeller of uniform pitch.

The pitch, from the analogy to the screw thread, is defined as the distance axially between two successive turns of the same helicoidal surface, or the distance which the screw would propel the boat providing it were supported on a corresponding immovable helicoidal surface, as that of a nut of equal pitch and suitable form.

The faces of many propellers, however, are so formed that the pitch is not the same at all parts of the surface. That is, the entire blade face does not belong to one helicoidal surface, as defined above, and the actual blade face could not be generated by a line in the manner indicated above. In some cases the faces of such variable-pitch propellers may be generated by a line and in others they cannot be so formed. The simplest general way in which to consider the face of a propeller of variable pitch is as a combination of a series of helicoidal elements, each of which belongs to some surface as above defined, and for each of which there is a definite pitch as resulting from such definition. The most common variation of pitch is axial or a gradual increase from the forward or leading edge of the blade to the after or following edge. In addition, variation from the hub to the tip is often found with or without increase axially, as above noted.

There seems to be no experimental evidence justifying the use of a variable pitch, and, so far as experience can show, the propeller of uniform pitch may be depended upon as the equal of any fanciful variation of pitch which has yet been suggested or employed.

The shape of the blade should be oval or well rounded at the tip. If the diameter has been reduced below the values indicated by the equations given in the preceding section, the form may be filled in or widened somewhat at the tips, but in all cases the corners should be well rounded off, with as nearly as may be a smooth and continuously-curved contour of blade.

BLADE AREA.

The area of the circle $\pi d^2 \div 4$, that is, the area of the circle swept by the tips of the blades, is commonly called the disk area. The actual blade area is usually proportioned to the disk area, and, in accordance with the rules given for four-bladed propellers in the preceding section, its amount is usually from .35 to .45 of such area. For a three-bladed wheel its amount will usually be found from .27 to .33 of the disk area, while

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for a two-bladed wheel the fraction will vary from .20 to .25. In other words, each blade, as a rule, may be given from .09 to about .12 of the disk area, the larger blades being the more permissible on the two- and three-bladed wheels. Experiment indicates that there is little or no use in continuing the addition of area beyond a certain point. With propellers of low pitch ratio, such as those from 1.0 to 1.2, no advantage can be looked for from an area beyond .40 to .45 of the disk area. As the pitch ratio rises, however, more and more area may be added with the expectation of some slight gain, but in no ordinary case will it be desirable to go much beyond .45 or .48 of the disk area.

MATERIALS.

Cast iron, ordinary brass, and bronze are all employed for propellers, as well as cast steel for those of large size. Cast iron is the cheapest, but it is relatively weak and brittle and the thickness of the section must be relatively large, and the edges cannot be made as thin as with brass or bronze. Brass gives a smoother surface and sharper edges, but does not allow any material reduction in the thickness. Bronze of high quality is the best material for propeller blades, giving a smooth surface, sharp thin edge, and reduced thickness, all of which are favorable to good efficiency. Of the various high-grade bronzes manganese bronze is usually preferred, and may be considered as the most desirable material for all high-grade work.

For an estimate of the actual thickness of the blade at the root, or where it joins the hub, we may use the formula:

$$t = A\sqrt{\frac{H}{b\,N\,n}}$$

where t = thickness in inches at root of blade.
H = horse power.
b = breadth of blade in inches at root.
N = revolutions per minute.
n = number of blades.
A = 13 to 16 for bronze and
18 to 20 for cast-iron or common brass.

Thus, for illustration, suppose $H = 8$, $N = 380$, $n = 4$, and $b = 4$. Then we have for cast iron or common brass, taking $A = 18$,

$$t = 18\sqrt{\frac{8}{4 \times 380 \times 4}} = .65 \text{ inch.}$$

For bronze we might have similarly

$$t = 15\sqrt{\frac{8}{4 \times 380 \times 4}} = .54 \text{ inch.}$$

REVERSING THE PROPELLER OF A MOTOR BOAT.

When the propeller is driven by a steam engine the steam is controlled by some form of reversing valve gear through which the engine is reversed without difficulty. With the usual type of gasoline engine, however, the operation, as elsewhere seen, is entirely different in character, and such an engine cannot be reversed in the same way as a steam engine.

In some cases, especially in small sizes, the sparking device may be so adjusted as to admit of starting the engine in either direction once it is stopped. In such cases the engine may, therefore, be reversed, and a propeller wheel of the usual type may be employed.

In the more usual case, however, the engine is intended to run in one direction only, and for reversing the propeller some means other than reversing the engine must be provided. To this end two means are available: A reversing mechanism is placed on the shaft, or the propeller itself is of the reversible type.

While the principle involved may be worked out in a variety of ways, it consists essentially of a bundle of gear wheels and friction clutches so connected that when thrown into the go-ahead position the whole mechanism is locked and acts as a form of rigid coupling connecting the engine with the propeller shaft, which therefore turns with the engine in the usual manner. When, on the other hand, it is thrown into the backing position the connection from the engine shaft to the propeller shaft is through the gearing, and in such way that the motion is reversed, so that in this way the engine may be allowed to turn continuously in one direction, while by means of a control lever the clutch may be thrown into the ahead or backing position, or, if desired, into an intermediate position where the engine shaft revolves freely without attachment to the propeller shaft, which, therefore, remains at rest.

In the case of the reversing propeller the shaft is continuous, or at least may be so, and the reverse is brought about by changing the obliquity of the blades of the propeller. The blades are each cast separate and mounted on the hub on a pivot in the center line of the blade. A lug or pin or some portion of the blade then extends into the hub, where it is attached to a rod carried through the hollow propeller shaft or otherwise to a control lever located in a convenient position for operation at the engine. In order to avoid complication such propellers are rarely made with more than two blades, though in one form three blades have been employed.

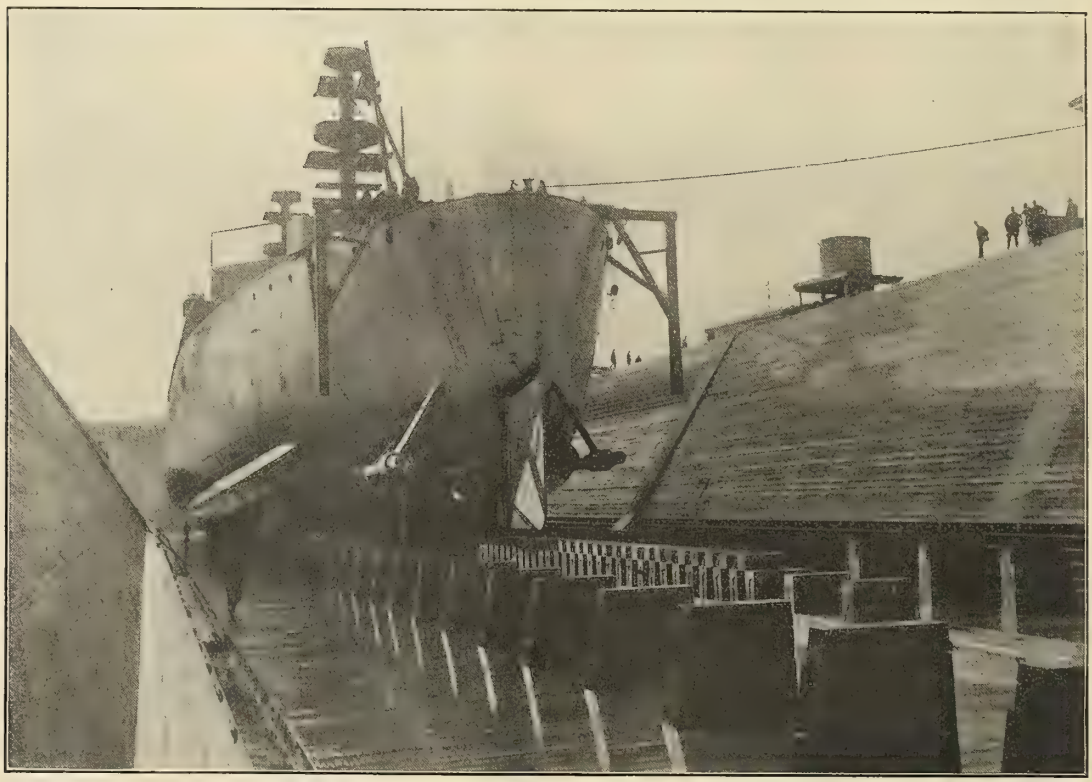
With this device the blades may be held in their usual position for go ahead, or thrown over to the opposite position for backing, or held in an intermediate position where the effective pitch of the blades is nothing, and where the boat will remain at rest or slowly forge ahead or back as the angle may be inclined one way or the other.

In regard to the change of pitch in this manner, it must be noted, however, that only in the one position for which the blades are originally intended will the faces lie in a true helicoidal surface. In all other positions the pitch will vary over the face of the blade; and while there will be some resultant or effective value, still operation at this setting will be at a sacrifice of efficiency. Such propellers should, therefore, be so designed and proportioned as to allow of operation for normal conditions in the setting which gives the true helicoidal surface for which the blades are primarily designed.

Regarding the choice of propeller it is probably fair to say that where propulsive efficiency is of importance a four-blade solid wheel may be recommended, and this with the usual type of gasoline engine will require a reverse mechanism in the shaft. In many cases, however, with boats of this type, the highest attainable propulsive efficiency is a matter of less importance than compactness or handiness of manipulation, and from these points of view some may prefer the reversing propeller to the reversing clutch. Either type of arrange-

ment provides a satisfactory mechanical solution of the problem of reversing the motion of the boat, and each is well represented by several different types of such mechanism in use in the practical field.

It is not expected that these remarks on propeller design for small boats will enable the amateur to confidently undertake all cases of such design with which he may come in contact, but they should serve at least to give indications of the lines along which his studies must be directed, and will be found, if intelligently applied, to give reasonably satisfactory results in all ordinary cases. An attempt to introduce a more detailed analysis of the problem, or to provide a method of design which should include in greater detail the influence of the various factors, would have resulted in methods unsuited to the purposes for which these chapters have been prepared.



STERN VIEW OF BATTLESHIP OHIO IN HUNTER'S POINT DRY-DOCK.

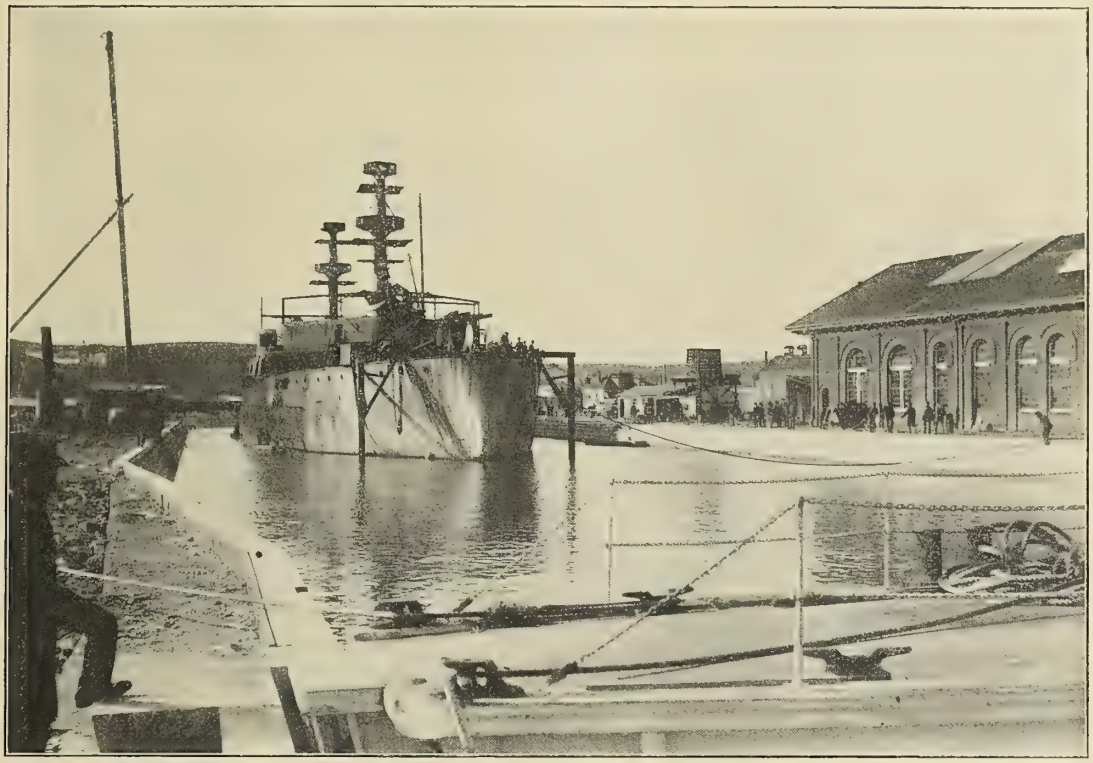
The Battleship Ohio in Hunter's Point Dry-Dock.

Two years ago the San Francisco Dry-Dock Company began the construction of a dry-dock at Hunter's Point, San Francisco harbor. The dock was completed in January of this year and was formally opened on January 29 to receive the hull of the United States battleship *Ohio,* which was launched by the late President McKinley, and has been under construction for nearly three years at the Union Iron Works.

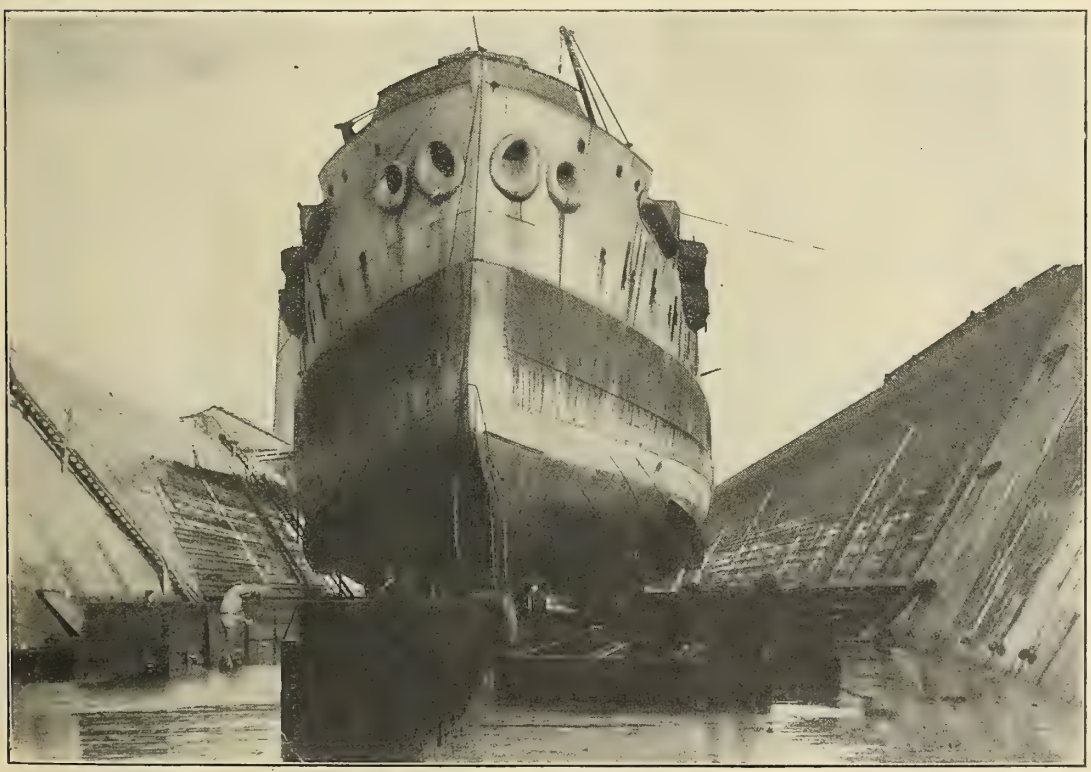
The new dock was described in Marine Engineering of August, 1902, and, it will be remembered, is of the following dimensions: 750 feet long, 122 feet wide at top and sloping down to 80 feet wide at the level of the top of keel blocks. At high tide there are 28 feet of water over the sill, and thus the largest vessel afloat could be dry-docked comfortably in this basin. The *Ohio* is 93 feet long by 72.2 feet beam, but at the time of docking drew only 17 feet of water with a displacement of 8,000 tons. She was towed from her moorings at the Union Iron Works by the tugs *Sea Rover, Sea Fox,* and *Sea Queen* and brought to Hunter's Point; at 11.40 A.M. the great caisson was floated away from the entrance of the dock and the battleship was warped inside the gate. The gate was then floated back, sunk in place, and at 12.45 P.M. the three centrifugal pumps, each with a capacity of 40,000 gallons of water per minute, started to discharge the water from the dock. At 2.50 P.M. the battleship was resting on the keel blocks and the dock clear of water. As the receding water exposed the hull of the vessel, an army of laborers attacked the barnacles and sea plants on the ship's bottom, and by the time all the water was out of the dock the hull was clean. Though the ship had not been docked for ten months, her bottom was not very foul.

As she entered the dock there were 11 feet of water between her keel and the sill of the dock; as she lay in the dock after the water had been pumped out, there were four feet to spare on each side of her at the narrowest part of the dock; and between her stern and the gate there was an unoccupied space of 357 feet.

Steamer Tionesta.—The Erie and Western Transportation Company's new steamship *Tionesta,* which will go into commission in the Anchor Line fleet this spring, was launched on December 15 from the Wyandotte yards of the American Shipbuilding Company. The vessel is 360 feet long over all and has a cargo capacity of 3,500 tons and passenger accommodations for 350.

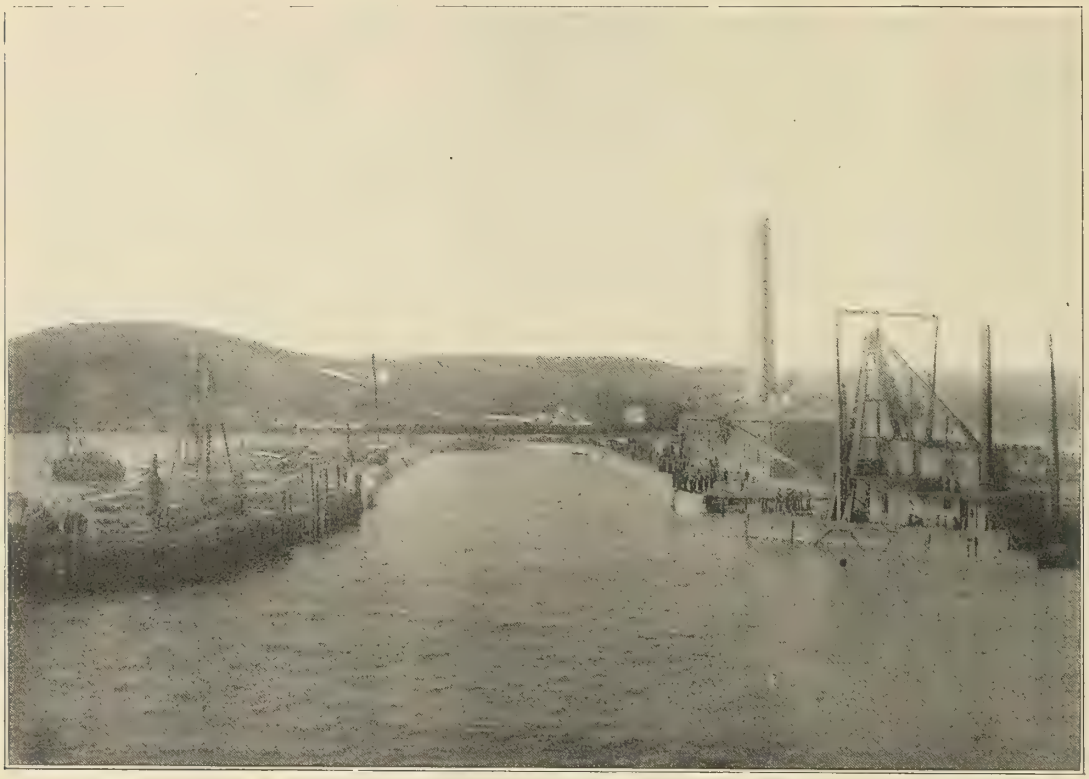


DOCKING THE BATTLESHIP OHIO AT HUNTER'S POINT.

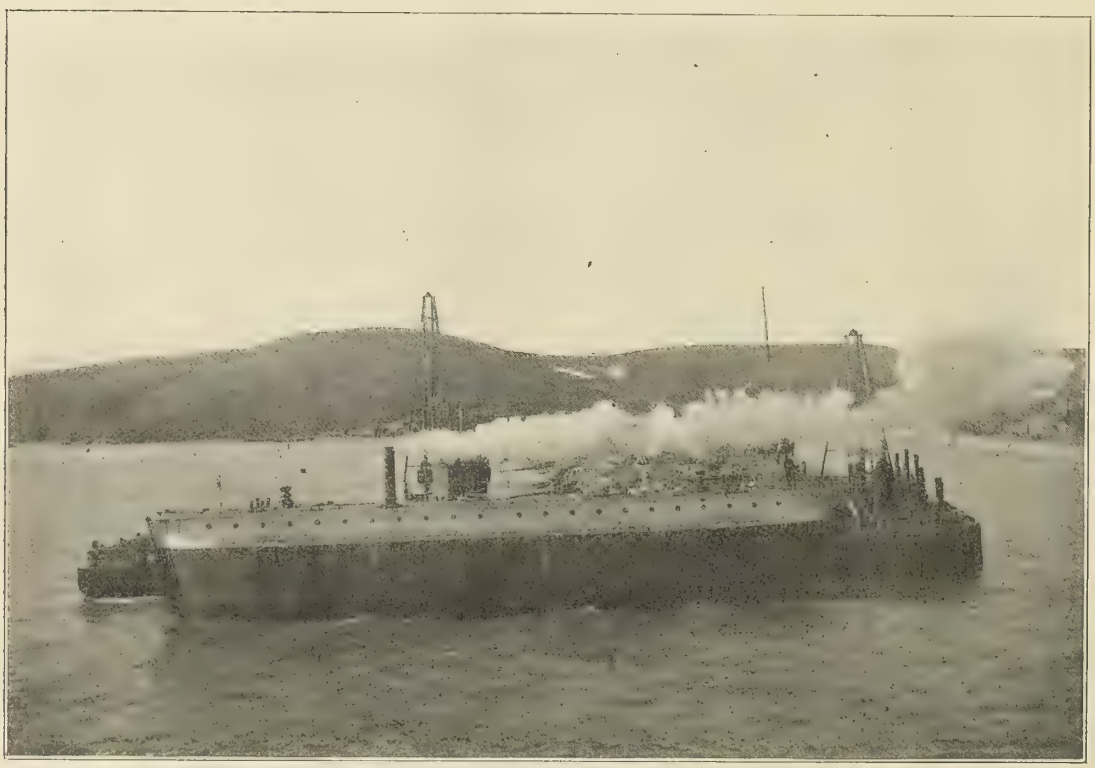


BOW OF THE BATTLESHIP OHIO.

[Photographed by P. F. Dana.]



NEW DRY-DOCK AT HUNTER'S POINT, SAN FRANCISCO HARBOR.



STEEL CAISSON OF THE HUNTER'S POINT DRY-DOCK.

Safety Valves.

BY H. K. SPENCER, U. S. R. C. S.

There are numbers of different rules for determining the size of safety valve necessary for a given boiler, and it is to be expected that there would be some difference in the results obtained, even though they are all good and general in their application, but there ought not to be the wide difference there actually is. It may be that some of the rules are not published as those who devised them intended they should be, and some may have been devised for certain conditions only, but, when no conditions are stated, we assume the rules to be general.

A safety valve, to fulfill its purpose, must be of a size sufficient to discharge all the steam made by the boiler when steaming at its full capacity, and it need be no larger. The area through which the steam is discharged is a cylindrical one, and is the product of the circumference of the valve and its lift; so the discharging capacity of a valve varies as its circumference, not its area, that is, as the diameter, not the square of the diameter. This is true until the valve has been raised to a height equal to one-fourth of its diameter, when the cylindrical area equals the area of the valve, after which, no matter how high the valve is raised, the discharging capacity remains constant and is measured by the area of the valve.

Thus the area of valve $= 1\text{-}4\,\pi d^2$, circumference of valve $= \pi d$, and when the lift is made equal to $1\text{-}4d$ the area of the cylindrical opening is $\pi d \times 1\text{-}4d = 1\text{-}4\,\pi d^2$.

The velocity of escaping steam is practically constant at all ordinary pressures, so the weight of steam discharged through a given orifice in any given time depends upon the pressure only because the density varies as the pressure. It is for this reason that safety valves are designed for the pressure at which the boiler will be worked when old. In what follows, it will be assumed that each pound of coal burned on the grates will evaporate 8 pounds of water, a figure undoubtedly large, The rate of combustion for natural draft will be determined by the rule $c = 3\sqrt{H}$, where c = coal in pounds per hour burned per square foot of grate, and H = height of smokestack in feet above lowest grates.

The discharging capacity necessary for a safety valve is expressed by the equation,

$$W = w \times A,$$

where W is the total weight of water evaporated per hour, and is consequently equal to the weight of steam which the safety valve must be able to discharge in one hour;

w is the weight of steam, in pounds, discharged per hour per square inch of orifice, through an opening similar to a safety opening, and is obtained from the equation, $w = 45 \times P$, P being the absolute steam pressure in pounds;

A is the cylindrical area in square inches, which is the actual area through which the steam is discharged.

The following additional notation will be used:

A_1 = area of safety valve in square inches;
d = diameter of valve in inches;
p = gage pressure in pounds;
H.S. = heating surface in square feet;
G.S. = grate surface in square feet;
C = total coal burned per hour in pounds and $= c \times$ G.S.;
a = area of high-pressure piston in square inches;
V = velocity of piston in feet per minute.

Some of the published rules are applied in two cases where the writer has the necessary boiler data. The boilers will be designated as No. 1 and No. 2. Boiler No. 1 is comparatively old, having been built about eighteen years ago; it is now worked at a pressure of 65 pounds, and is fitted with one spring-loaded safety valve. No. 2 is new, and though at present its working pressure is 160 pounds, it is assumed that the boiler will be worked at a pressure of 140 pounds when old, and the necessary safety-valve area is calculated for that pressure. This boiler is fitted with two safety valves on a single base. Both are marine boilers of the ordinary Scotch type. The boiler data are:

	No. 1.	No. 2.
Heating surface, sq. ft.	1,700	1,300
Grate surface, sq. ft.	63	45.5
Height of stack, ft.	45	54
Working pressure, lbs.	65	160
Diameter high-pressure piston, in.	26	25
Piston speed, ft. per minute	600	800
Number of safety valves	1	2
Diameter of each valve, in.	5	2.75

The size of safety valve given for No. 2 boiler is that for one of the two valves with which it is fitted, and, except in one case, is that of a valve of half the capacity of the one obtained by rule. The diameter of valve is given for convenience of comparison..

The lift of the valve, in any case, may be called L.

No.	Authority.	Rule.	Boiler. No. 1.	Boiler. No. 2.
1	Rankine	$A = \frac{\text{H. S.}}{3p}$	6.71 ins.	2.98 ins.
2	Rankine	$A_1 = .006\ W$	8.78 "	5.53 "
3	U. S. Inspectors	$A_1 = \frac{\text{G. S.}}{3}$	5.17 "	3.11 "
4	Prussian Law	$A_1 = \frac{36\ \text{G. S.}}{p}$	6.66 "	2.73 "
5	Philadelphia Inspection Law,	$A_1 = \frac{2.25\ \text{G. S.}}{p+8.62}$	4.96 "	2.10 "
6	Unknown	$A_1 = \frac{.5\ W}{p+10}$	9.26 "	4.12 "
7	Inst. Eng's and Shipbuilders in Scotland. 2 valves size found by this rule.	$A_1 = \frac{.6\ \text{H. S.}}{p}$	6.32 " *	2.66 "
8	Hutton's Steam Boiler Construction.	$A_1 = \frac{4\ \text{G. S.}}{p}$	6.31 "	3.13 "
9	Hutton's Steam Boiler Construction.	$A_1 = \frac{W}{40\sqrt{p}}$	6.31 "	3.26 "
10	Bourne	$A_1 = \frac{aV}{300\ P}$	4.12 "	2.32 "
11	Unknown	$A_1 = \frac{C}{30}$	7.32 "	4.61 "

* Size of single valve equal to combined area of two valves obtained by this rule.

When using rule No. 1, L is supposed to be at least equal to $\frac{d}{20}$, and in rule No. 4, at least equal to $\frac{d}{36}$.

An inspection of the results tabulated above shows the extent of the variation, and also that the rule which gives the largest valve for boiler No. 1 does not give

the largest size for No. 2; so also for the smaller sizes. Applying the formula, $w = 45\ P$, it gives for boiler No. 1, $w = 45 \times 80 = 3{,}600$ pounds per hour per square inch of safety-valve opening, as the rate at which the steam would be discharged from that boiler.

For boiler No. 2 the rate is 6,975 pounds per hour per square inch of safety opening.

By the law governing the Steamboat Inspection Service of the United States, it is required that "each spring-loaded safety valve shall be supplied with a lever that will raise the valve from its seat to a distance of not less than that equal to one-eighth of the diameter of the valve opening," so, in calculating the maximum discharging capacity of a safety valve, its lift may be taken as $\frac{d}{8}$. The lift, L, of any safety valve, due to the action of the escaping steam, even when blowing off with the boiler steaming at its full power, is generally considered to be about $\frac{d}{24}$, so that will be taken as the lift of the valve when discharging automatically, and in calculating what can be termed the minimum rate of discharge.

	No. 1.	No. 2.
Maximum discharging capacity largest in pounds steam per hour when $L = \frac{d}{8}$	121,200	167,500
Rate in pounds coal per hour per sq. ft. grate at which boiler can be fired and largest valve discharge all steam made, with $L = \frac{d}{8}$	240	460
Maximum discharging capacity largest valve in pounds per hour, $L = \frac{d}{24}$	40,400	55,820
Rate in pounds coal per hour per sq. ft. grate at which boiler can be fired and largest valve dispose of all steam made, $L = \frac{d}{24}$	80	153
Maximum discharging capacity smallest in pounds steam per hour, $L = \frac{d}{8}$	23,990	21,260
Rate in pounds coal per hour per sq. ft. grate at which boiler can be fired and smallest valve dispose of all steam made, $L = \frac{d}{8}$	47.6	58.4
Minimum discharging capacity smallest valve in pounds steam per hour, $L = \frac{d}{24}$	7,980	7,090
Rate in pounds coal per hour per sq. ft. grate at which boiler can be fired and smallest valve dispose of all steam, $L = \frac{d}{24}$	15 8	19.5
Discharging capacity actual valve in pounds steam per hour, $L = \frac{d}{8}$	35,350	41,440
Combustion rate permissible, actual valve open $\frac{d}{8}$, in pounds coal per hour	70.4	113.8
Discharging capacity actual valve in pounds steam per hour, $L = \frac{d}{24}$	11,780	13,800
Combustion rate permissible, actual valve open $\frac{d}{24}$, in pounds coal per sq. ft. grate	23 5	37.9

Rule 6. Rule 2. Rule 10. Rule 5.

It is seen from an inspection of the above figures that the largest valves are very much in excess of any possible requirement, even when discharging automatically, while the smallest valves, when discharging automatically, are sufficient only for moderate rates of combustion. The actual valves seem to be about what is required, a little large, perhaps, for both boilers. No. 1 is always worked under natural draft, while No. 2 is sometimes worked under moderate forced draft.

The size of the safety valve should be determined only from the amount of steam the boiler can produce when fired with the greatest amount of coal that can be burned on the grates. With natural draft the maximum combustion rate is about 25 pounds coal per hour per square foot of grate, and with forced draft 50 pounds is about the limit, torpedo boats excepted. It should be understood that the higher the rate of combustion the lower the rate of evaporation; the figure, 8 pounds water evaporated per pound of coal, is good work for a moderate rate of combustion, and it would be nearer the fact to say that, with forced draft, when burning 40 to 50 pounds of coal per hour per square foot of grate, 6 pounds of water are evaporated per pound of coal. It is seen that the rate of combustion is most variable, and that the rate of evaporation is also variable in a less degree, so the best way to approximate the generating power of the boiler is by means of the maximum I.H.P. of the engine. The steam consumption of any ordinary type of engine can be predicted with fair accuracy; 17 pounds per I.H.P. per hour for triple-expansion engines, 19 pounds for compounds, and 25 pounds for simple condensing engines can be called average figures. The maximum quantity of steam required of a boiler furnishing steam to a triple-expansion engine, then, is 17 × I.H.P., which quantity of steam the safety valve must be capable of discharging; that is, the product of the area of the actual valve opening, A, and the rate in pounds per hour per square inch of valve opening at which the steam is discharged, must equal 17 × I.H.P., or,

$$A \times 45\ P = 17 \times I.H.P.$$

Assuming the lift of the valve to be $\frac{d}{24}$,

$A = \frac{\pi d^2}{24}$; now $A_1 = \frac{\pi d^2}{4}$, so $A = \frac{A_1}{6}$, and then,

$A = \frac{2.27\ \text{I. H. P.}}{P}$ for the area of the safety valve for a boiler furnishing steam to a triple-expansion engine.

In the same way, for a boiler furnishing steam for a compound,

$$A = \frac{2.54\ \text{I. H. P.}}{P}$$

For one furnishing steam to a simple engine,

$$A = \frac{3\ 33\ \text{I. H. P.}}{P}$$

Valves over 5 inches in diameter are seldom employed, especially with high pressures. The necessary area should be obtained, if a valve over 5 inches is found to be required, by using two or more valves of equivalent area.

Monthly Shipbuilding Returns.—The Bureau of Navigation reports 53 vessels of 3,234 gross tons built in the United States and officially numbered during the month of January, 1903.

THE VIBRATIONS OF STEAMSHIPS, WITH SPECIAL REFERENCE TO THOSE OF SECOND AND HIGHER PERIOD.—II.*

BY REAR ADMIRAL GEO. W. MELVILLE, ENGINEER-IN-CHIEF, U. S. NAVY.

Are the Unbalanced Forces of Higher Period in the Four-Crank Engine Important?

BALANCE OF DEUTSCHLAND

I shall answer this question by an examination of the unbalanced forces and moments of the *Deutschland* as she is, and as she would be if the cranks were at right angles. I thereby, incidentally, examine more fully than I did last year the claim of the Schlick engine to be called "balanced." In my last year's paper I showed that the *Deutschland* at ninety revolutions had an unbalanced second-period moment of 1,910 foot tons. The quotation just given from Mohr also shows, very impressively, the magnitude of this moment.

On the other hand, turning to Fig. 25 of Mr. Taylor's paper of last year, showing a complete scheme of the setting of the Schlick engine and the corresponding values, for each setting, of the second-period forces and moments, the result for what he pronounces the best settings is *apparently* most satisfactory. The unit he has used is stated in his Appendix C, and is the inertia of the whole reciprocating mass of the forward crank taken about the after crank. Now, this unit is enormous, and necessarily dwarfs anything that could be tolerated on board a full-powered ship.

The YST system was brought forward as an improvement on the ordinary four-crank engine, which is admitted by all to be very unbalanced. To what extent does this system effect an improvement? That, I think, is the fairest statement of the case.

Mr. Taylor's specification of the best conditions of setting of the YST engine, and the actual setting of the *Deutschland,* are as follows:

Mr. Taylor's specification, sections 27 and 28 of his paper:

Crank 4, reciprocating mass = 1
" 3, " " = about 1.5
" 2, " " = somewhere between 1 and 2
" 1, " " = about 1
" 4, crank angle zero
" 3, " " approximately 135° to 160°
" 2, " " in third puadrant somewhat less than 270°
" 1, " " approximately from 445° to 60°

Forward pair of cylinders to be as close as practicable, and after pair as close as practicable.

DEUTSCHLAND.

See p. 104 and Figs. A and D of my paper.

Crank	I	22.2†	= 1.
"	II	32 ÷ 22.2	= 1.44
"	III	32 ÷ 22.2	= 1.44
"	IV	22.3 ÷ 22.2	= 1.00
"	I	crank angle zero	
"	II	"	158° 10′
"	III	"	265° 10′
"	IV	"	63° 20′

*Read at the meeting of the Society of Naval Architects and Marine Engineers, New York, November, 1902.

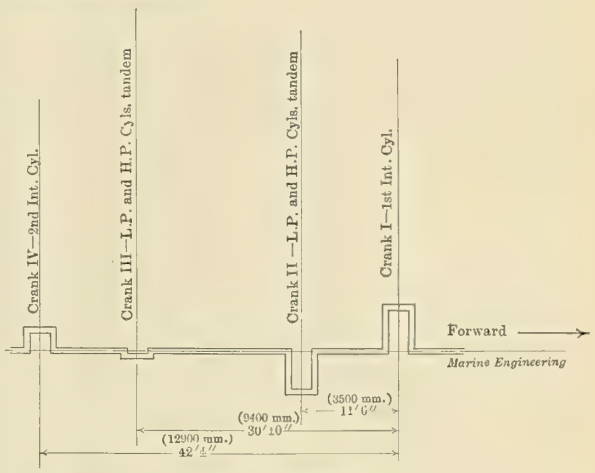
†From Fig. A cranks II and III have inertia = 268 tons for $m = 32$ tons. Therefore, for cranks I and IV we have $m = \frac{186 \times 32}{268} = 22.2$, and $m = \frac{186.6 \times 32}{268} = 22.3$ tons respectively.

Distance between cranks I and II, and between cranks III and IV = 11 feet 6 inches. Distance between cranks II and III = 19 feet 4 inches.

(Mr. Taylor's crank 4 and my crank I correspond, both being forward.)

Hence we see that, according to Mr. Taylor's specification, the setting of the *Deutschland* is almost ideal. For the sake of clearness, I reproduce Figs. A to D of my last year's paper, which give the examination for balance and unbalance of the *Deutschland* as made. Let us compare, then, the *Deutschland's* engine as it stands with what it would have been had cranks I and II been opposite, cranks III and IV opposite, but the first pair at right angles to the second pair, as in the usual arrangement.

First take the moving masses exactly as they are in the *Deutschland.* The result for first period is shown in Fig. 17. The forces and moments are the same as in Fig. A, and we obviously get the unbalanced force = $\{(268-186)^2 + (268-186.6)^2\}^{\frac{1}{2}} = 115.5$ tons; and the resultant unbalanced moment of 3,103 foot tons is similarly calculated; or, they may be at once measured from the diagram.



CRANK CENTERS. Fig. D

Fig. 18 gives the results for second period.

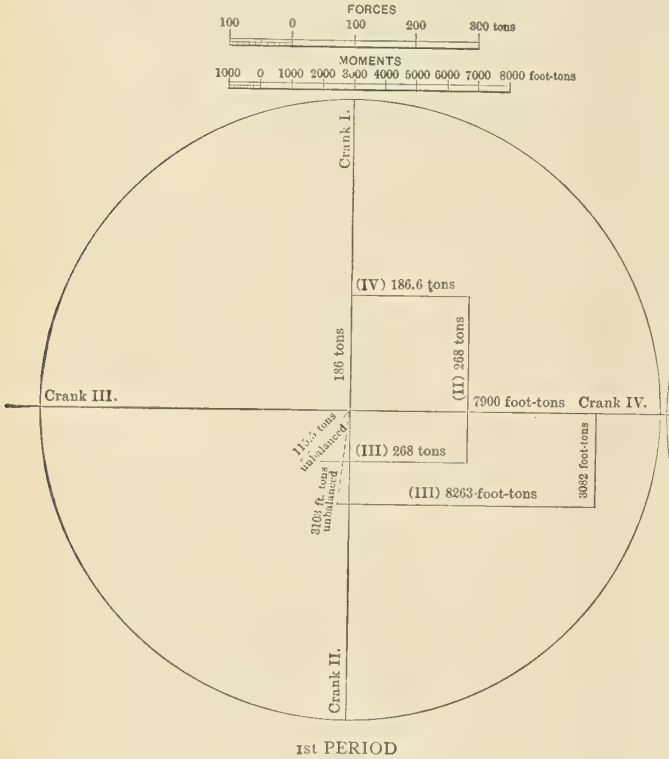
For fourth period, a diagram like Fig. C would show all the cranks in one plane, similarly directed. Hence the whole resultant is a vertical force passing through the center of the engine. This resultant is, obviously, from Fig. C, = .48 + .69 + .69 + .48 = 2.34 tons.

If, in Fig. C, the resultant had similarly been taken through the center of length of the engine, the value of the unbalanced moment would have been 8.8 tons. This is found by producing the line of the 1.71 unbalanced force through the center of the circle, laying off on this line a moment of 1.71 × 21 1-6 = 36.20 foot tons,—21 1-6 feet being the half length between cranks I and IV,—and combining it with the 37.2 foot tons. The new and old force systems are, of course, equivalent, but the smaller moment is given in the subjoined table to make the comparison fairer. If we treated the first-period moments in the same way, instead of 3,103, 2,848, and 1,736, in Table A, we would get 1,913, 2,090, 1,253 foot tons, respectively; values which would help greatly to strengthen my arguments. This would be a perfectly proper proceeding, as the engine is practically symmetrical about its center—but I feel that I can

safely dispense with the help this would give. The large second-period unbalanced couples would scarcely be affected by a change in the point round which the moments are taken, as there is almost no unbalanced force.

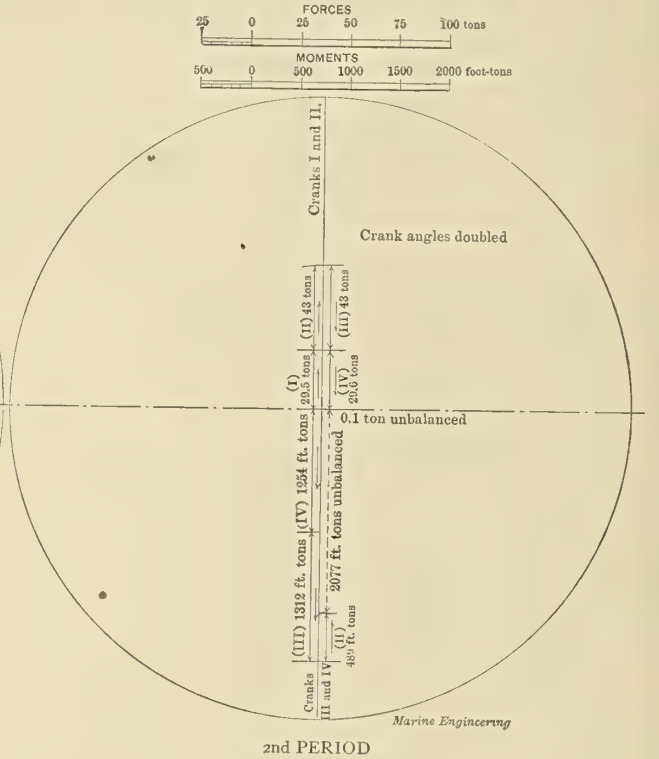
But considerable weight is added to the H.P., first I.P., and L.P. pistons to produce the YST balance. This we may deduct as follows:

	Weight as made	Reduction.	Reduced Weight.
Crank I	22.2	1	21.2 tons
Crank II	32	2 6	29.4 "
Crank III	32	2 6	29.4 "
Crank IV	22.3	0	22.3 "



1st PERIOD

Fig. 17



2nd PERIOD

Fig. 18

Calculating from these reduced masses, we get—

Unbalanced force of first period = 90.3 tons.

Unbalanced moment of first period = 2,848 foot tons.

But further, we may, by comparatively small revolving balance weights at the ends of the engine, or by counterbalancing the cranks, exactly balance all revolving weights. This leaves only the reciprocating masses to be dealt with. These I have taken in my paper as 5-8 of the total moving masses for each crank, and from this should be deducted the metal that may be saved off the pistons. We thus have—

Crank I	22.2 × 5-8 − 1	= 12.9 tons.
Crank II	32 × 5-8 − 2.6	= 17.4 "
Crank III	32 × 5-8 − 2.6	= 17.4 "
Crank IV	22.3 × 5 8	= 13.9 "

We can at once find the first-period forces and moments for each of these masses. Multiplication by .2540 and .0041 will give their forces and moments of second and fourth period, respectively, which are to be combined as in the foregoing diagrams. The resultants are given in Table A.

The last column shows what can readily be effected with the ordinary arrangement of cranks. Overbalancing the horizontal forces by slightly increasing the revolving counterbalances, which but infinitesimally increases the total machinery weight, would very considerably improve this result without producing sensible horizontal vibrations. By taking the resultant at the middle of the engine, the moment 1,736 would have been reduced to 1,253 foot tons and the comparison made fairer. But, taking the figures as they stand, we have a residual first-period force of 48.4 tons, or less than 1-5 of the force from crank II of the *Deutschland* as made. This would be considered a very small amount of unbalance. Even the 115.5 tons in the third last column is much less than half the force from crank II. We also see that the moment 1,736, which has been balanced by the Schlick system, is considerably less than the second-period moment of 1,976 which has been left practically unaffected.

But it may be objected that the third last column is the fair comparison. Accepting this position, I will show that the first-period moment of 3,103 foot tons (or more properly 1,913 foot tons about center of the engine) is much the less serious than the 2,077 foot tons of second period that the YST system fails to reduce materially; that is, under equally favorable conditions for producing vibration the second-period moment would produce *very much* the greater effect. All this will be evident whenever Table B has been studied.

TABLE A.

UNBALANCED FORCES AND MOMENTS OF DEUTSCHLAND AS MADE, AND WITH CRANKS AS IN FIGURE 17.
FORCES IN TONS, MOMENTS IN FOOT TONS, REVOLUTIONS 90 PER MINUTE, SUPPOSED.

Period.	Force or Moment.	As Made from Figs. A, B, and C.	Cranks as in Fig. 17.		
			Pistons as Made.	Pistons Reduced.	Piston Reduced and Revolving Weights Balanced.
1st	Force	0	115.5	90.3	48.4
	Moment	0	3103	2848	1736
2d	Force	1.7	0.1	2.1	2.1
	Moment	1910	2077	1976	1976
4th	Force	1.71	2.34	2.12	2.12
	Moment	8.8*	0	0	0

A glance at Table A should, at least, effectually dispose of the belief that second-period moments are negligible for the YST engine. The only way of justifying such belief would be by showing that the energy of second period is very quickly damped. I will give, in the last section, some of the results of an examination of this important subject. The question at the head of this section regarding the importance of higher-period forces in the four-crank engine must be answered with the most emphatic affirmative.

Are the Forces of Higher Period Sufficient to Account for Vibrations of Corresponding Period?

OUTLINE OF INVESTIGATION.

In Appendix I of my paper for *Engineering* I discuss, much more fully than would be convenient here, the question of the response of the ship to forces of various periods. I will only indicate briefly the line of reasoning and state the most important result.

I suppose only one period of vibration present at one time, as the problem becomes immensely complicated on any other supposition. Supposing the engine foundation immovable, take $F \cos n\theta$ as the expression for the inertia force of nth period, from equation (5) below, which the engine would apply as it revolved. But, as the ship is an elastic structure, the force of this period would cause a response at the point of application $= a \cos (\theta - \alpha)$. θ is the crank angle and α is the angular epoch of the response, the value of which varies as we approach or recede from perfect synchronism. For perfect synchronism, $\alpha = \frac{\pi}{2}$ which shows that the response lags one-quarter period behind the exciting force. If we follow this motion carefully, it at once becomes evident that energy will be communicated to the ship every vibration. It can readily be shown that the measure of this energy, per vibration, is πFa.

This supply of energy will cause a to increase, and this increase would go on until the ship is ruptured, were there not some cause of absorption of energy at work. It is analogous to the filling of a reservoir. However large the reservoir and however small the inflowing stream, it will be filled in time if there is no outlet. The rise of water level will only cease when the rates of inflow and outflow are equal. It is therefore essentially wrong to suppose, as has so often been done, that because F is very small for fourth and higher periods, it may be rejected. Before doing this it is necessary to examine carefully the whole question of the transfer of energy.

The absorption of energy in the vibrating ship is principally due to the action of the water. The resistance to motion may be taken as very approximately proportional to the square of the velocity. This law only I suppose here. I further suppose a nodal vibration, though Berling's observations seem to indicate that, for shorter vibrations than those with two nodes, the mode of vibration becomes very complex and cannot be compared to the nodal vibration of a thin rod. The particular supposition I make as to the number of nodes for the various periods, is:

Period	1	2	4	6
Number of nodes	2	3	4	5

This is to get definite numerical results; but the theory which I bring forward does not depend on a special set of suppositions.

The linear response will be greatest midway between the nodes, hence the unbalanced resultant *force* of the engine will produce its greatest effect if applied there. The greatest angular response will be at, or very near, a node, and the unbalanced *moment* of the engine will produce its greatest effect if applied at this point. For the sake of comparison of the various periods I suppose the forces applied midway between the nodes and the moments at the nodes, though this cannot be done with one and the same engine. As before, "force × frequency" is the measure of the importance of a vibration.

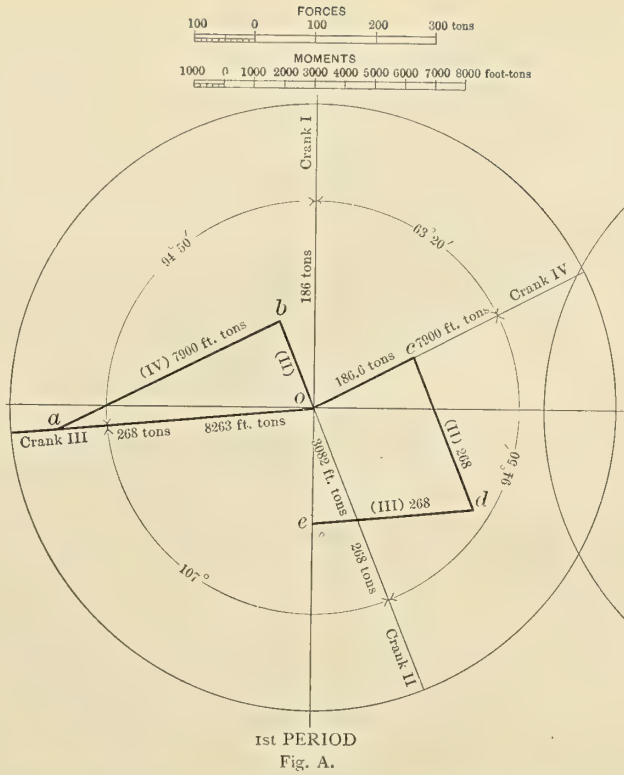
On the foregoing supposition the results in Table B can be demonstrated. In the second column are shown the forces which would produce vibrations of equal importance for the various periods. In the third column the same comparison is made for the moments. The two columns have no relation to one another. In the last column I give the coefficients for the various periods from the following equation for the inertia of a mass M moving with the crosshead:

Inertia form

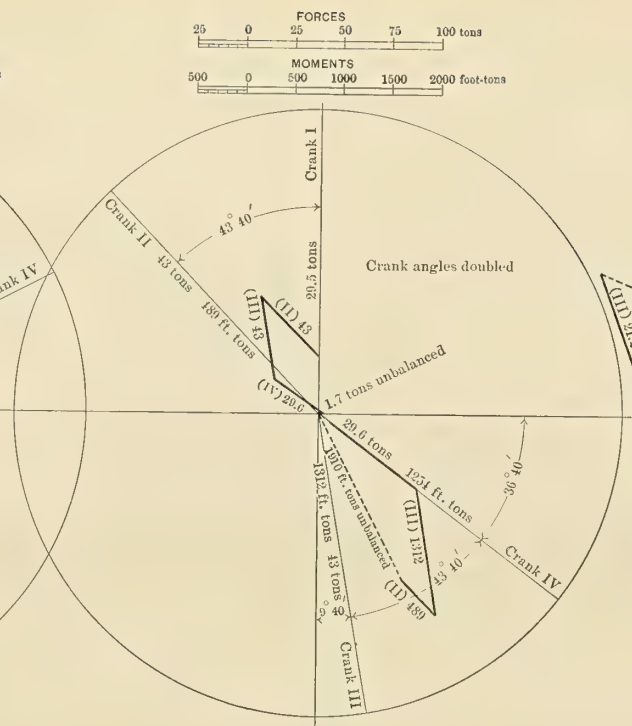
$$= \frac{M\omega^2 R}{g} (\cos\theta + .2540 \cos 2\theta - .0041 \cos 4\theta + .00007 \cos 6\theta + \text{etc.}), \quad (5)$$

where ω is the angular velocity of the crank in radians per second, supposed uniform; R, the crank radius in feet; g, the intensity of gravity; θ, the angle which the crank makes with its top stroke position, the engine being vertical inverted and having a connecting rod

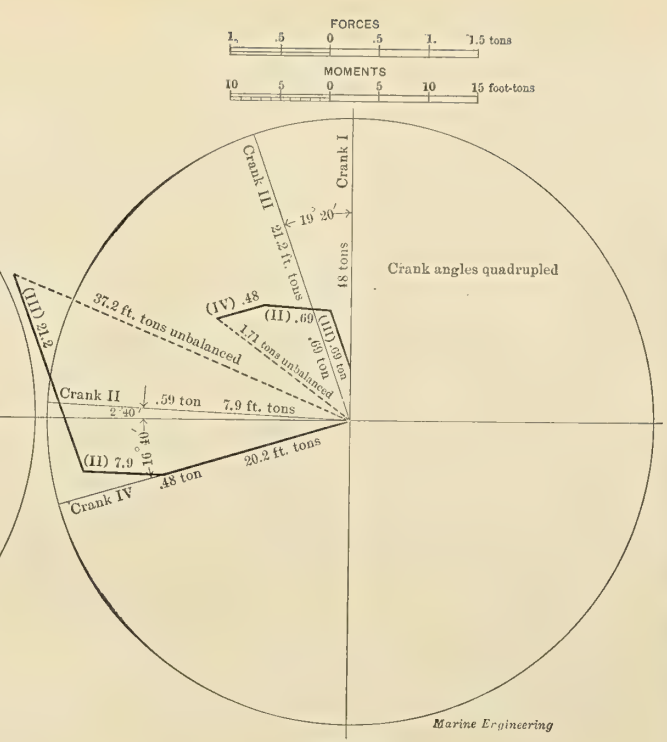
*This figure is reduced from that of Figure C for reasons noted above.



1st PERIOD
Fig. A.



2nd PERIOD
Fig. B.



4th PERIOD
Fig. C.

four cranks in length. All terms but the first, in the foregoing series, on the right hand of equation (5) arise from the finite length of the connecting rod.

TABLE B.*

RATIO OF FORCES AND MOMENTS OF EQUAL IMPORTANCE.

Period.	Resistance Proportional to 2d Power of Velocity.		Coefficients of Equation (1).
	Forces.	Moments.	
1st	1	1	1
2d	.0625	0415	.2540
4th	.0039	.0020	.0041
6th	.00077	.00031	.00007

* To make this table perfectly intelligible, I will show very simply that, under the suppositions made, the forces $F_1 = 1$ of 1st period, and $F_2 = .0625$ of 2d period, will give equally important vibrations.

If for 1st period a vibration having $a_1 = 1$ is produced by F_1, we must show that F_2 will produce a 2d-period vibration having $a_2 = 1\text{-}8$. For the force produced on a body of mass m is $\frac{mn^2\omega^2 a}{g}$, and force × frequency $= \frac{mn^3\omega^2 a}{g}$, where n is the number of the period and ω the angular velocity of the engine crank shaft. Taking in turn the values,

1st period $n = 1$ $a_1 = 1$
2d period $n = 2$ $a_2 = 1\text{-}8$

we get

1st period $\frac{m\omega^2}{g} \times 1^3 \times 1 = \frac{m\omega^2}{g}$

2d period $\frac{m\omega^2}{g} \times 2^3 \times 1\text{-}8 = \frac{m\omega^2}{g}$

Thus the values of a_1 and a_2 supposed, belong to equally important 1st and 2d period vibrations.

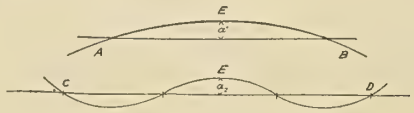
It remains to show that if for 1st period the rates of communication and absorption of energy are equal they will be equal for 2d period also.

Rate of communication of energy per vibration $= \pi Fa$. Thus,

Per vibration, for 1st period $= \pi \times 1 \times 1$.

" " " 2d period $= \pi \times 1\text{-}16 \times 1\text{-}8 = \frac{\pi}{128}$.

In considering the rate of absorption of energy by the water we have the ship vibrating in different modes for 1st and 2d period.



TEACHING OF TABLE B.

The intensity of the forces and moments of the various periods which arise in any engine will, of course, depend on the arrangement of cranks and other points in the design. But we see how liable we are to get a response of second period if the circumstances are favorable. Hence, whether we take the smallest or largest value of the first-period moment in Table A, we at once see that, in the YST system, the moment balanced is of *far* less importance than the second-period moment which *cannot* be balanced.

Such are the figures deducible on the assumptions I have fully stated. To those who have not carefully studied the question, it will at first appear incredible that the moments of first and second period, which, under equal circumstances, are of equal importance, are in a proportion in any way approaching 24 to 1, as Table B shows. We cannot escape from the conclusion that the latter are of first-class importance by supposing a more favorable law of resistance. Even with a resistance varying according to the first power of the velocity—a law much more potent in damping out higher periods than we can possibly suppose—the ratio of first to second period moments of equal importance is 6 to 1. Even here, then, for equally favorable conditions, the second period would be far the more annoying.

Further, I maintain that my conclusions *are* borne out by the facts, and that the higher-period vibrations are *far* the more objectionable. This I am about to enforce. Table B indicates a similar conclusion as to fourth-period vibrations, and the frequent occurrence of these bears strong testimony to the truth of the whole deductions. For were the coefficients under the headings of Forces and Moments much too small, we would conclude that those of fourth period would never appear. Even sixth period, with a favorable arrangement of cranks, has a chance of putting in an appearance; and it no doubt composes part of the general tremors of the ship, which are so irritating, though these may be due to jars, or, as Macalpine and Flood pointed out in 1894,* to the resistance not following the simple harmonic law.

CONCLUSIONS SUPPORTED BY PALLOGRAMS FROM KAISERIN AUGUSTA.

The pallograms from the *Kaiserin Augusta* bear strong evidence to the truth of Table B. Fig. 10 shows the largest first-period vibration for this ship given by Berling.

Probably here we have nearly perfect synchronism, as no doubt he was investigating both the worst and best conditions. Similarly Fig. 11 gives, with probably one exception, the worst second-period vibration, and we may again suppose synchronism.

TABLE C.

COMPARISON OF IMPORTANCE OF 1ST AND 2D PERIOD VIBRATIONS AND INTENSITY OF FORCES AND MOMENTS FOR KAISERIN AUGUSTA.

Figure.	Number of Engines Working.	Revolutions.	V.	Relative Maximum Value of Moments and Forces.	
				Revolving Parts Balanced.	Revolving Parts Unbalanced.
10	3	102	.67	1.	—
11	2	105	1.05	$\frac{1\times.2540\times105^2\times2}{102^2\times3}=.179$	$.179\times\frac{5}{8}=.112$

The connecting rod I have taken as four cranks in length, which seems to be indicated by a very small scale drawing in the translation of von Jaski's paper.† In this form of engine, supposing only the reciprocating masses unbalanced, the unbalanced forces and moments of first and second period are, as is well known, in the proportion of the first- and second-period forces, equation (5). Hence the factor .2540. If the revolving parts are unbalanced, and the reciprocating parts form 5-8 of the whole moving masses, we must multiply the .179 in the second last column by 5-8 to get the correct relative values of first- and second-period moments. The factors $105^2\times2$ and $102^2\times3$ correct for the revolution and number of engines at work.

Here we have, then, a relatively very small force and moment of second period producing a second-period vibration over 50 per cent. more important than that of first period.

In judging the figures in Table C it should clearly be borne in mind that, with resistance varying as the square of the velocity, the second-period forces or moments required to give $V=1.05$ are much larger than those which would have been required to give $V=.67$ for this period; they are, in fact, larger in about the proportion $1.05^2/.67^2=2.46$. Thus the support which is given by the pallograms of the *Kaiserin Augusta* to the truth of Table C is very striking.

There being first- and second-period vibrations present

Suppose for first period it vibrates in any mode, say as at *AEB*, and for 2d period as in any other mode *CED*. The resistance is supposed proportional to the square of the velocity, and therefore varies from point to point of the ship.

But the flexure between any contiguous pair of nodes produces practically a similar curve whatever the number of nodes. (This no doubt holds more closely for a thin rod than for a ship, and the flexure outside the outer nodes is not similar to that of the parts between; both of which slightly modify what immediately follows.) Hence whatever the number of nodes, for comparative purposes we may use the velocity as *E*, midway between two nodes, when estimating resistance, and thus avoid a difficult integration over the whole surface of the ship or rod.

The engine is supposed placed at the point *E* always between two nodes, when considering forces, as there they have the greatest effect.

The resistance is $C\times n^2\omega^2a^2$, and the energy absorbed per vibration $K\times n^2\omega^2a^3$, where C and K are approximately constants. Thus, energy absorbed

Per vibration for 1st period $=K\omega^2\times1^2\times1^3=K\omega^2$.

" " " 2d " $=K\omega^2\times2^2\times\frac{1}{8^3}=\frac{K\omega^2}{128}$.

Hence, if the rates of communication and absorption of energy are equal for 1st period, they are equal for 2d, so that $F_1=1$ and $F_2=1\text{-}16$ give equally important vibrations under the conditions supposed.

*See *Engineering*, vol. lviii., p. 83.

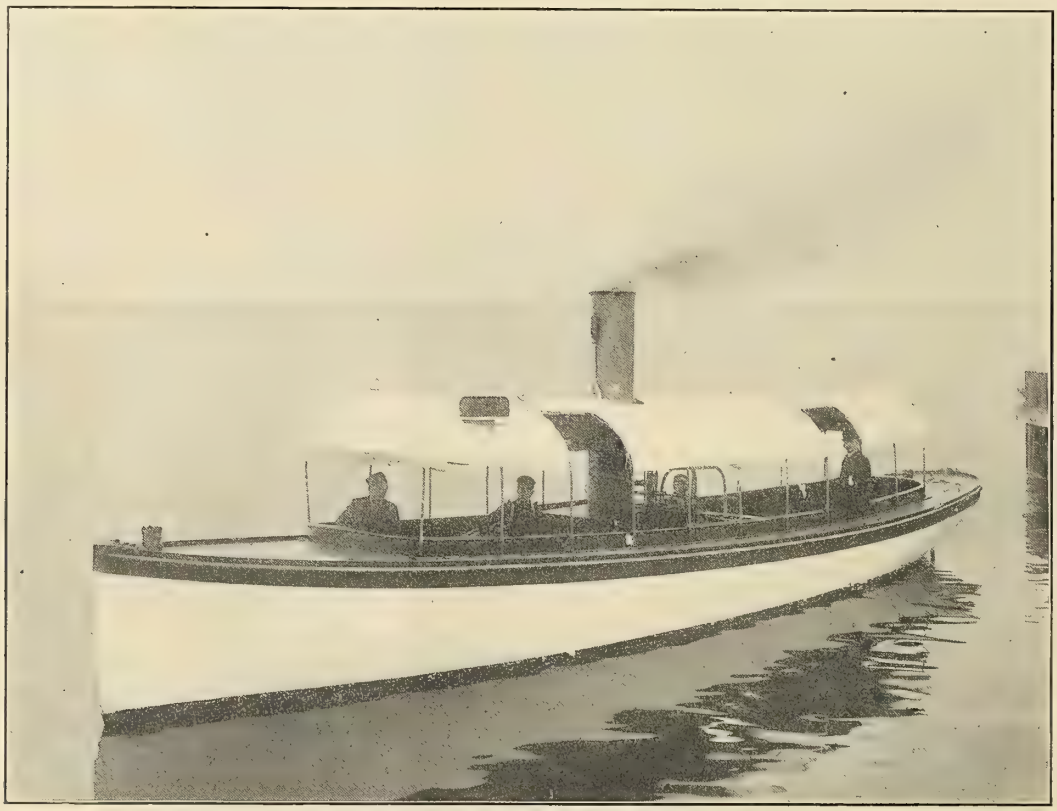
†Results of Investigations of Recent Engines of the German Navy, by Chief Constructor Kohn von Jaski. *Schiffbau*, June 8, 1900, p. 507. Translation, *Journal American Society Naval Engineers*, vol. xiv., p. 18.

would be sure, in this case, to produce a powerful damping effect on the latter, so that the case is even stronger than appears from the figures. More exact comparison would be unprofitable, as the data are somewhat doubtful; we do not know that synchronism was exact or that the relative positions among the nodes for first and second period were equally favorable. Besides, the vibrations are not measured at the same part of the ship. But it practically assures us that the rate of absorption of energy varies with a power of the velocity much over the first, and probably as high as the second.

Theory, then, leads to the conclusion that the higher-period unbalanced forces of the engine are able to produce the vibrations of corresponding period found by experiment.

STEAM LAUNCHES FOR THE WAR DEPARTMENT.

Two 45-foot steam launches were recently finished by the Electric Launch Company, Bayonne, N. J., for the War Department of the United States Government. One of the launches, named the *Lancaster,* was shipped to New Orleans and the other one is for use at New London, Conn. The vessels were built from the same specifications, which called for very heavy construction throughout. They are much the same as the Navy Department launches. The successful bidders have for many years been building electric and gas-engine launches, sailing yachts, and other small craft. The boats here described were built within seventy-five days of the date of contract.



STEAM LAUNCH LANCASTER, BUILT FOR THE U. S. WAR DEPARTMENT.

TORSIONAL VIBRATION OF SHAFTING.

The question of torsional vibration of the shafting has come much to the front in the last few years. It is a subject of the deepest interest and importance. I will not enter on it in this paper further than a few words. In only a limited number of cases can exact balance of the impulses causing torsional vibration be effected; hence its presence in some degree we must be prepared for. If it is not to produce a greatly aggravated transverse vibration of the ship, we must adopt a balanced engine. But as torsional vibration is almost always of second, third, or fourth period, any engine balanced only for first period cannot, in this case, be considered to be balanced at all.

I believe that it will be admitted that I have now supplied overwhelming evidence of the correctness of the position I took up last year.

HULL CONSTRUCTION.

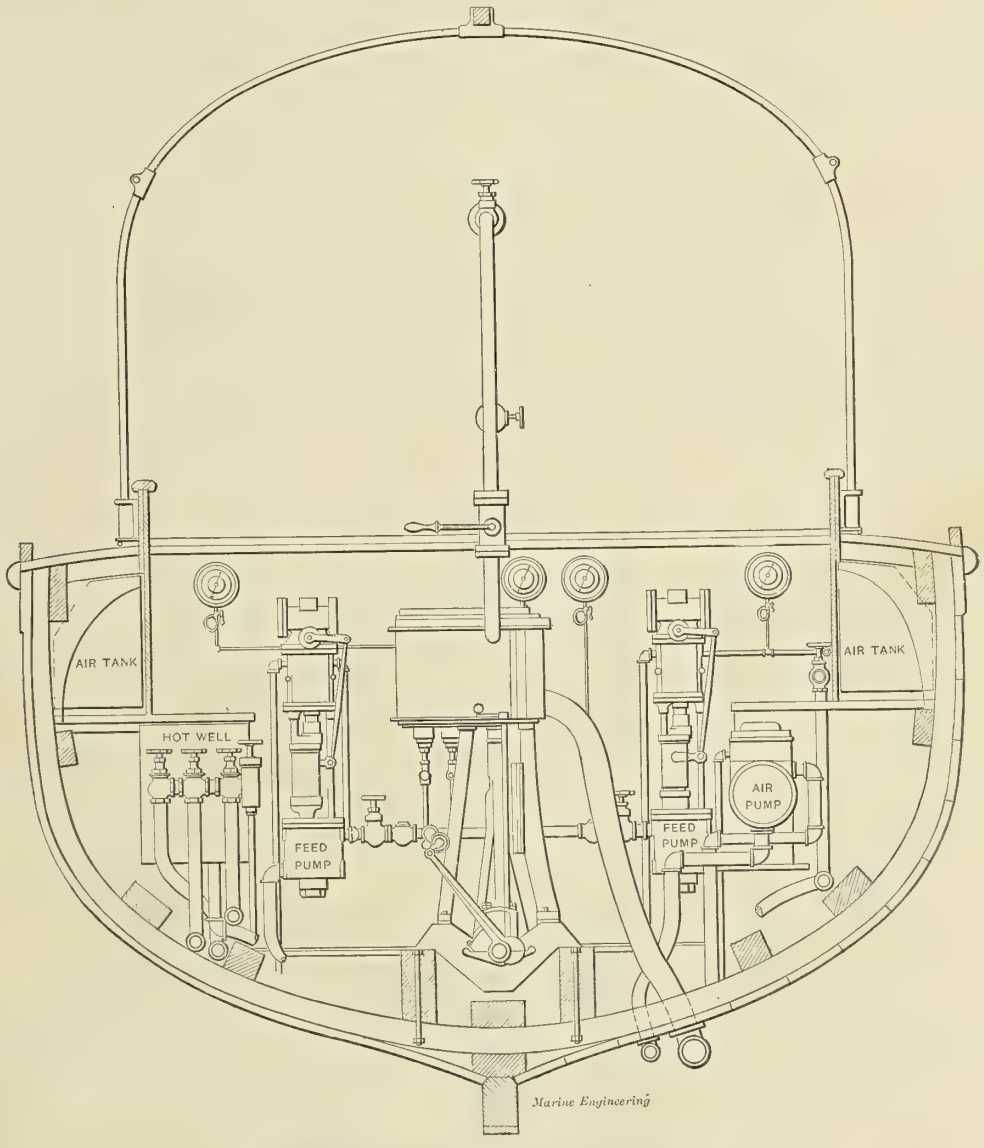
Keels of clear tough oak, 4 by 6 inches, with a 6 by 2-inch covering piece and a 4 by 1-inch oak shoe or false keel. The frames are of clear white oak, steam bent, 1 1-2 inches sided by 2 inches molded at heads and 3 inches at heels. The keelson is laid on the frames above the keel and is of 3 by 6-inch yellow pine. The stem and stern post of oak of size and shape as shown. The floors are of oak, 1 3-4 by 3 inches at throat. The side keelsons are yellow pine, 3 1-3 inches, and the bilge stringers, 2 1-2 by 5 inches, made of the same material and tapered at the ends. The clamps are tapered, of yellow pine, 2 by 6 inches. Three breast hooks of oak, natural knees, 3 1-2 inches thick. Knees were shown of oak or hackmatack. Sheer strake of oak, 1 1-2 by 7 inches, tapered. Outside planking of clear white cedar, 1 inch thick and not over 7 1-2 inches wide amidships. Fender 1 1-2 by 3

inches, of oak and tapered fore and aft. Rail of oak, 1 1-2 by 2 inches. Plankshеer of oak, 4 1-2 by 1 inch, tapered at ends. Deck of clear white pine, 1 3-4 inches width by 1 inch thick. Coamings around cockpits to be of oak, 1-2 inch thick by 6 inches high, with a cap of cherry or mahogany 2 5-8 by 1 1-2 inches. Deck beams and floor beams of oak, 1 1-2 by 2 inches, dovetailed into clamp and stringers. Deck stringer of oak, 2 by 2 inches. Staving of cockpits to be of 5-8 by 3-inch tongue and groove mahogany. Cockpit floors to be of 3 by 3-4-inch white pine tongue and groove ash gratings where shown. Filling pieces of white pine, 6 by 3 inches. Ceiling of sides of hull in coal bunkers of 1-2-inch tongue and groove yellow pine, top of coal bunkers of 7-8-inch ash. Engine keelson of yellow pine, about 4 by 12 inches, as may be required, cut down over the floors 1-2 inch. Stern seats to be of ash with coaming around edges for cushions, with lockers underneath. Forward seats to be of ash with lockers underneath; all materials to be of suitable dimensions.

Keel, keelson, stringers, clamps, covering piece, and shoe to be in one piece for their entire length, as shown.

Fastenings.—Heavy copper fastenings and rivets shall be used for planking and frame. All screws of joiner work shall be of brass. All exposed nail or screw heads to be countersunk and flush plugged.

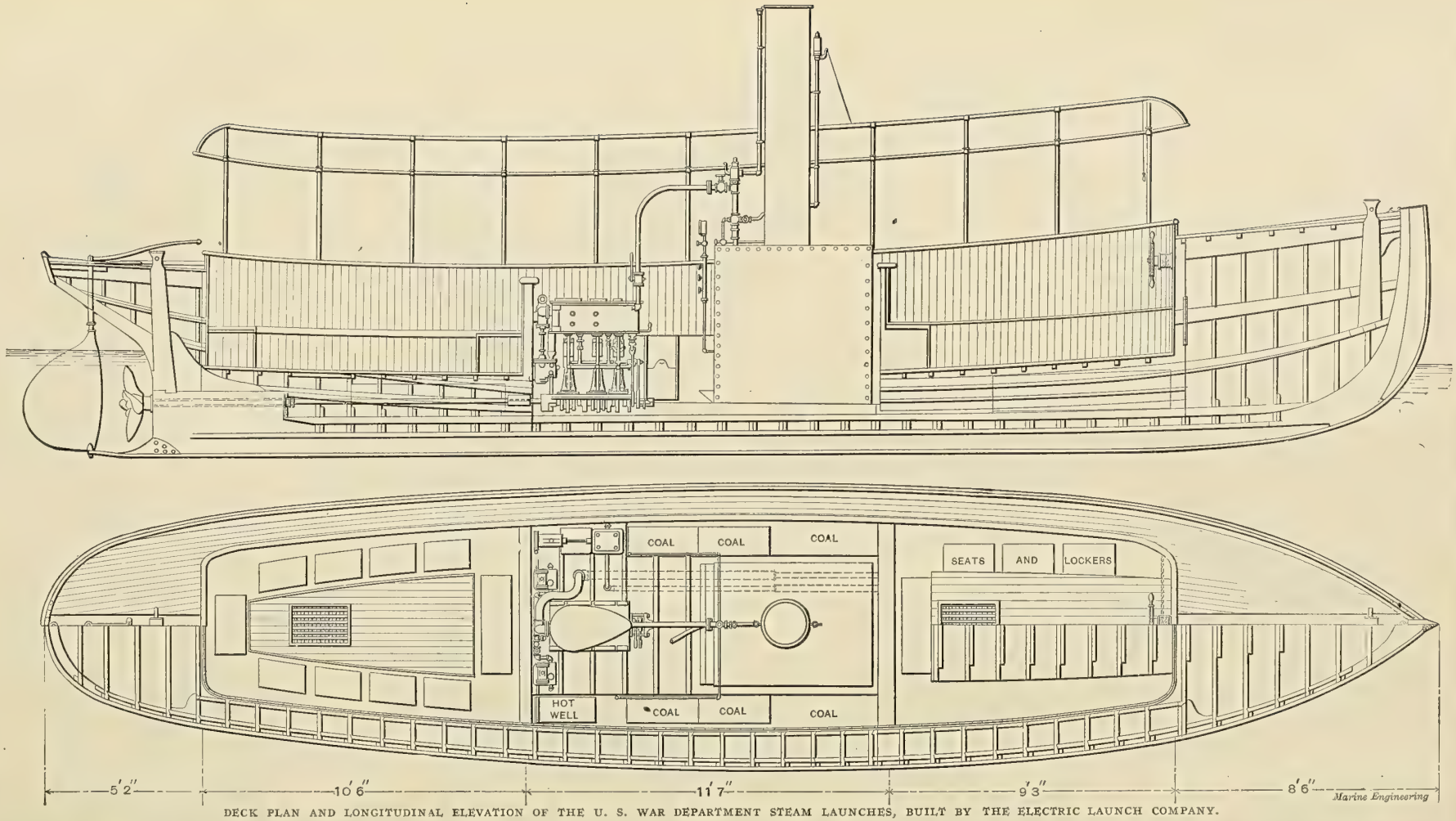
Calking.—Seams of planking and decks to be calked with spun cotton and payed with paint and filled with white lead putty.



MIDSHIP SECTION OF WAR DEPARTMENT STEAM LAUNCHES.

Painting.—Planking and frame to have three coats of best white lead outside and inside, except floors and inside of hull in machinery space, which is to be finished with three coats of lead color paint. Lettering on bow.

Coppering.—Launch to be coppered over with felt with 14-ounce metal nailed with 5-8-inch nails; copper to extend from a line 6 inches above load-water line forward and 3 inches aft on sides around keel for the full length of the boat. The keel shoe to be put on outside of the copper, in order to protect it from being torn should the boat ground.



DECK PLAN AND LONGITUDINAL ELEVATION OF THE U. S. WAR DEPARTMENT STEAM LAUNCHES, BUILT BY THE ELECTRIC LAUNCH COMPANY.

Varnishing.—Fender, planksheer, decks, and interior to have three coats of best spar varnish, after being properly filled with an approved filler and rubbed down.

Towing Bitts.—Stern post to extend through the deck, as shown, and to have a strong composition pin through same. A riding bitt with composition pin to be strongly framed into fore part of launch.

Rudder, Scag, etc.—The rudder and scag, propeller wheel, stern bearing, rudder post, stuffing box, cutwater or steam band, and all other outboard fittings to be of bronze and of approved design.

Bulkheads.—Water-tight bulkheads, of 1-8-inch galvanized steel, to be placed forward and aft of cockpit, an air chamber being arranged on each end. Manhole to forward chambers. Copper air tank about 3 feet long, of No. 18 gage, and as large as is possible, properly stiffened, to extend along the entire side under deck, as shown. Steel bulkheads athwartships, one forward of boiler and one aft of engine, the latter to have opening with cover, where necessary, to allow of getting at low-pressure valve gear and valves as may be re-

quired. Cover to be hinged and bolted in place; joints to be water-tight.

Coal Bunkers.—To be constructed of 1-8-inch galvanized steel and angles alongside of boilers, substantially as shown. Provide openings in top for filling and steel sliding chute on side to fire room.

Lockers.—To be arranged under the seats of forward and after cockpits and at the side of the engine room, all as shown. All fittings of polished brass.

MACHINERY.

Engines.—The engine is of the "Sullivan" make, of the fore-and-aft triple-expansion type, capable of working under a steam pressure of 250 pounds without undue strain, supported on steel stanchions well braced and secured, and thrust bearing attached. All bearing surfaces are extra large, well provided with oiling attachments, and have an arrangement for taking up the lost motion. A central sight feed oiling reservoir is connected to all bearings and pins by small brass tubes. The crank shafts are of high-grade forged steel, with cranks set at 120 degrees; propeller shaft is of Tobin bronze, and the crankpin and other main bearings to run in white metal boxes. All bolts and nuts are finished and case-hardened, with checks on the main bearings and moving parts. The necessary relief valves and drain cocks are attached and piped up as required.

NAME—Lancaster. NATIONALITY—American. TYPE—Steam launch.
BUILDER—Electric Launch Company. LAUNCHED—1902. OWNER—U. S. War Department.

Length over all	45 ft.	Valves	all piston
Beam	9 "	Steam pressure	225 lbs.
Depth	5 "	Revolutions per minute	495
Displacement, load	24,500 pounds	**Boiler**	one water-tube
Corresponding draft, bow	2 ft. 6 ins.	Size	floor space, 4 by 5 ft.
" " stern	3 " 6 "	H. S.	321.5 sq. ft.
Engines, type	triple	G. S.	12.68 " "
Cylinder diam.	4, 7, and 11 ins.	**Propeller**	4 blades, bronze casting
Stroke	7 1-2 ins.	Diameter	28 ins.

Boiler.—Steam is generated by a water-tube boiler, of latest and approved design, of sufficient capacity to drive the above engine at least 500 R.P.M. without excessive forcing and to supply all the steam required, the boiler being built in accordance with the laws of the United States Treasury Department, for a working pressure of 250 pounds, and proven tight under a hydrostatic pressure of 500 pounds per square inch. The ashpan is of galvanized steel; wrought-iron shaking grates; ashpan doors and fire door, as well as entire casing, to be of the best for their purpose and of approved materials and thickness. Handholes, steam hose and connections are fitted for blowing soot and ashes from water heaters and base of tubes or manifolds. The safety valve is set at 175 pounds. The boiler and heating coils are of standard 1-inch steam pipe. The boiler is fitted with all necessary gages and cocks. The gases from the fire are led into a double smokestack of No. 18 iron.

Pumps.—The air pump is of the Marsh type, 4 by 4 by 6 inches. The boiler is fed with two Davidson single-cylinder feed pumps, 4 by 4 1-2 by 4 inches. A hand pump and ejector are connected with the bilge.

Piping.—All the piping is of brass or copper. The main steam exhaust and feed pipes fitted with flanges brazed on according to navy practice. All the fittings are of the heavy pattern of the smaller types and have unions brazed on. The valves are of extra heavy composition.

Condenser,—An outward type condenser, capable of condensing the steam from the engine and maintaining a good vacuum, is of copper, the whole secured to the throat of the hull and protected by cutwater blocks.

Water Tanks.—There are three water tanks, made of No. 18 copper, well stiffened and secured and provided with filling pipes, suctions, strainers, water-tight handholes. One of these tanks is placed under the floor of the after cockpit, with a shaft casing running through the same, one under the floor of the forward cockpit and one in the air chamber at the bow, with a combined capacity of 250 gallons, and all leading to manifold in the engine room.

Miscellaneous.—The steering gear to consist of a 24-inch brass steering wheel forward and a hand tiller aft, connected up with brass wire rope and chains, as shown in the photograph. The awning is carried on a galvanized frame set in composition sockets and is of No. 5 canvas. The other parts of the outfit include screenboards for the side lights and bow lights, chocks, cleats, two flagpoles, three boathooks, one pair of ash oars, a set of brass sailing lights and engine-room lanterns, bells for engine room, a complete set of wrenches, oil cans and firing tools, a 75-pound galvanized folding anchor with 120 feet of manila 3-inch rope, fenders, and spare parts of the engine will include one set of grate bars, feed heating coils, two links and valve rods, a set of crankpin frames, a set of wristpin blocks, and one link block.

It is evident from the above specifications that these launches are especially heavily built and of the very best material throughout. The hull alone weighs about 9,000 pounds, and the copper fastenings, including air tanks, weigh from 1,300 to 1,500 pounds. The cost of

building under these specifications is about 10 per cent. above what it would be under the usual form of specifications.

The rigidity of these vessels was shown on their trial trips, which were run in smooth water on Newark bay. With the engines running at 500 revolutions there was no vibration perceptible. Steam was raised from cold water to 150 pounds pressure in about 25 minutes; on another day, with a wood fire, the steam pressure was raised in 12 minutes and the safety valve was set to blow off at 225 pounds. The steam jet was kept on slightly in order to maintain the pressure when running at full speed. One of the feed pumps sufficed to keep the boiler well filled with water, and the Marsh air pump gave a good vacuum when making 70 strokes per minute. The following data were obtained:

Steam pressure, pounds	175	225	250
1st receiver pressure, pounds		80	
2d receiver pressure, pounds		13	
Vacuum, inches		26	
Revolutions per minute	440	495	525
Mean speed, miles		10 9-10	

A full turn of the boat was made in about two lengths.

GAS ENGINES AND THEIR TROUBLES.—IX.

BY E. W. ROBERTS.

Making Repairs.

Quite often the occasion arises when the operator of a gas engine finds it necessary to make a repair when quite a distance from a machine shop. There are tricks in every trade, and the little trick of knowing what to do in an emergency of this kind is quite convenient. For instance, it would be apparently disastrous if the operator of the single-cylinder engine should find that his cylinder head packing had blown out and had no sheet asbestos on board. Again, sheet asbestos of the proper kind for packing a gas-engine cylinder head is not always readily obtainable, and hence the knowledge of a satisfactory substitute would be of great assistance under such circumstances. Sheet copper embedded in white lead is not only a good substitute for asbestos packing, but a great many people think it an improvement. Both the gasket and the joint should be well coated with white lead and the cylinder head drawn firmly into place by the nuts on the cylinder-head stud, taking another turn after the engine is warmed up. Sheet lead is another very good substitute for asbestos, and, in fact, has proven a satisfactory gasket in places where the asbestos packing would not hold. Even well-oiled manila paper will make a temporary repair, and for packing in places not exposed to high temperatures, especially for the cover of a rotary pump, oiled paper is about as good as anything else. It should be put on, however, in from four to six thicknesses and tightly compressed.

REGRINDING A VALVE.

For regrinding the exhaust or the inlet valve of a four-cycle engine the grit that is given off from a grindstone is by many considered much better than emery. Finely powdered and sifted glass, especially of the flint-glass variety, will also answer for this purpose, or even a little of the grit from a piece of fine glass paper, commonly known as "sandpaper," will also do the work. This glass paper grit, if not too coarse, is a very good material, as it is usually a good quality of flint glass. Valves should be ground with a continuous rotary movement and a slight pressure, always using the grit mixed with oil.

It often happens that when putting too great a strain on a nut the thread on either the bolt or in the nut is stripped and will not take hold. This quite often happens when the nut was originally a poor fit and there is little thread remaining. Sometimes you can make the nut hold temporarily by winding the bolt with a piece of string. If the nut be a small one it can be struck with a hammer on the flat, so as to make the hole elliptical, and this will make it fit temporarily, if the strain is not too great. This is not a very mechanical method, but it will sometimes answer in an emergency.

Leaks in the gasoline system may be temporarily mended with common brown soap, and a leak in the gasoline pipe may have a temporary repair by wrapping it with tape or with strips of cloth well covered with the same substance. Shellac, which is not soluble in gasoline, may be used in place of soap in conjunction with the tape. It should be put on while wrapping the tape around the pipe, as it dries very quickly, and afterward a good coating of shellac should be put over the entire joint. Both of these methods should be used for temporary repairs only and a permanent repair made with solder at the first opportunity.

In launches equipped with an air-pressure system for starting it is often quite difficult, when the outfit is new, to hold the air pressure for any length of time, but after the machinery has been exposed to the moisture of the air the joints will often rust tight. However, a very good way to insure a tight job at the outset is to take down this air system piping, tin all the threads, and screw them together again while hot with a little melted solder on each joint. It is not a bad plan to insist that the manufacturer use this method of construction when putting up the job, and thus save yourself the trouble.

ELECTRIC EQUIPMENT.

It sometimes happens that the battery will give out when in a locality where it is impossible to secure supplies for its renewal or new cells of a similar type. Frequently, however, it is possible to obtain dry cells such as are used for electric bells, and these will often answer for a short time, especially if two batteries of about five cells each be set up so that the current may be switched from one to the other at intervals of half an hour. These batteries will probably answer for several days or until new ones of the regular type can be obtained. If a bell battery of the wet type be the only sort obtainable, the cells should be fitted with wooden covers to prevent the electrolyte from splashing over. The small storage batteries used to operate a phonograph will do very well in an emergency.

The platinum tips or contact points on the igniter electrodes sometimes get loose and are difficult to secure when far from a city. In an emergency brass or silver will answer, but are not, of course, so good as platinum. It might as well be noted here that there are several alloys fully as good for this purpose as platinum, and at the same time considerably cheaper. In the absence of facilities for brazing, the points may be put entirely through the electrode and riveted.

Turning off the cylinder oil for a few minutes before the engine is stopped will, with many engines, allow the surplus oil to be burned away from off the cylinder walls and prevent a gummy deposit in the packing-ring grooves. Turning off the igniter switch before closing the throttle and stopping the engine will usually leave a charge in the cylinder at the last stroke of the piston, and make the engine easy to start next time you are ready to go. Never forget to turn off the igniter switch when stopping the engine, as it will save your battery and a possible short-circuit. In order to determine whether the spark coil is working properly, the simplest way is to hold a light piece of steel or iron near the end of the core so that it will swing. Then each time the circuit is closed and broken the piece of iron will swing toward the coil and be immediately released when the circuit is broken. This is a very simple method of finding just when ignition takes place in the cylinder.

In some makes of engines flat springs are employed on the igniter and they are very likely to break at the most inopportune moments. They can usually be replaced by a portion of an old clock spring, which, if not stiff enough in a single thickness, may be strengthened by using two thicknesses or more. If it is necessary to drill the spring it may be accomplished effectually by keeping the point of the drill sharp and using turpentine as a lubricant. A brass spring will also answer for a short time, and soft brass may be "tempered" by hammering it. This will, of course, thin the brass and necessitates the use of a sheet somewhat thicker than required for the spring. Generally a brass spring should be about twice as thick as a steel for the same purpose. German silver wire is quite often used for valve springs instead of steel, especially in locations where the spring is apt to get hot and steel would lose its temper. In case the valves hammer badly on their seats and make a disagreeable noise, the trouble may be overcome by the use of a light buffer spring or a piece of soft rubber or even leather.

RUNNING WITH ONE CYLINDER.

In case one of the cylinders on a multiple-cylinder engine should become inoperative while on a trip, the compression in the cylinder should be relieved, if possible, as this will allow the other engines to run at a higher speed. In the case of a damaged piston or connecting rod the rod should be disconnected from the crank pin and the piston blocked up or removed from the cylinder. In the case of a two-cycle engine, relieving the compression would not prevent the suction of air from the vaporizer, and in case a common vaporizer is used, so that the gasoline cannot be cut off from the cylinder, the passage from the crank case should be blocked in order to save gasoline and the driving of the explosive mixture into the muffler.

WATER-JACKET PUMP.

If the water pump stops working the trouble may be caused by clogging of the screen at the opening into the sea, or a small particle of dirt under one of the check valves, or too strong a spring on a spring check. The writer has cured a pump of failure to operate by removing the spring from the top of the suction check, the weight of the valve itself being found sufficient to bring it back to its seat. On high-speed engines with the piston pump operated directly from the crank shaft, it will often be found that the water will not follow the piston and the pump will simply draw a vacuum and no water. This is a trouble that I have seen cured by placing a flexible rubber bag just outside the suction check, either on the leg of a tee or so as to form part of the suction pipe. This insures a supply of water close to the pump, which will flow in at quite high speeds.

It should also be noted that a high-speed pump should be located, if possible, below the water line and that the suction pipe should be of ample proportions. Increasing the size of the suction pipe is sometimes an effective remedy for a badly-acting pump. A rotary gear pump is really to be preferred to one of the piston variety for all speeds, as the absence of check valves and the immunity from troubles caused by small particles of dirt are decided advantages. These pumps work best when the water flows to the suction, although when new they will generally draw quite a little distance.

In a two-cycle engine it is very important that the crank case be kept tight, for a leaky crank case will cause a considerable falling off in the power of the engine. Occasionally, although not very often, a leak in the wall of the crank case will develop from the jarring out of a particle of sand. These holes may be stopped effectually by means of a paste made as follows: Mix litharge, red lead, and glycerine in equal proportions so as to make the mixture about the consistence of putty. Apply this in the same manner as putty is used in filling wood. This paste should then be allowed to harden and it will be found almost as sound as the iron itself.

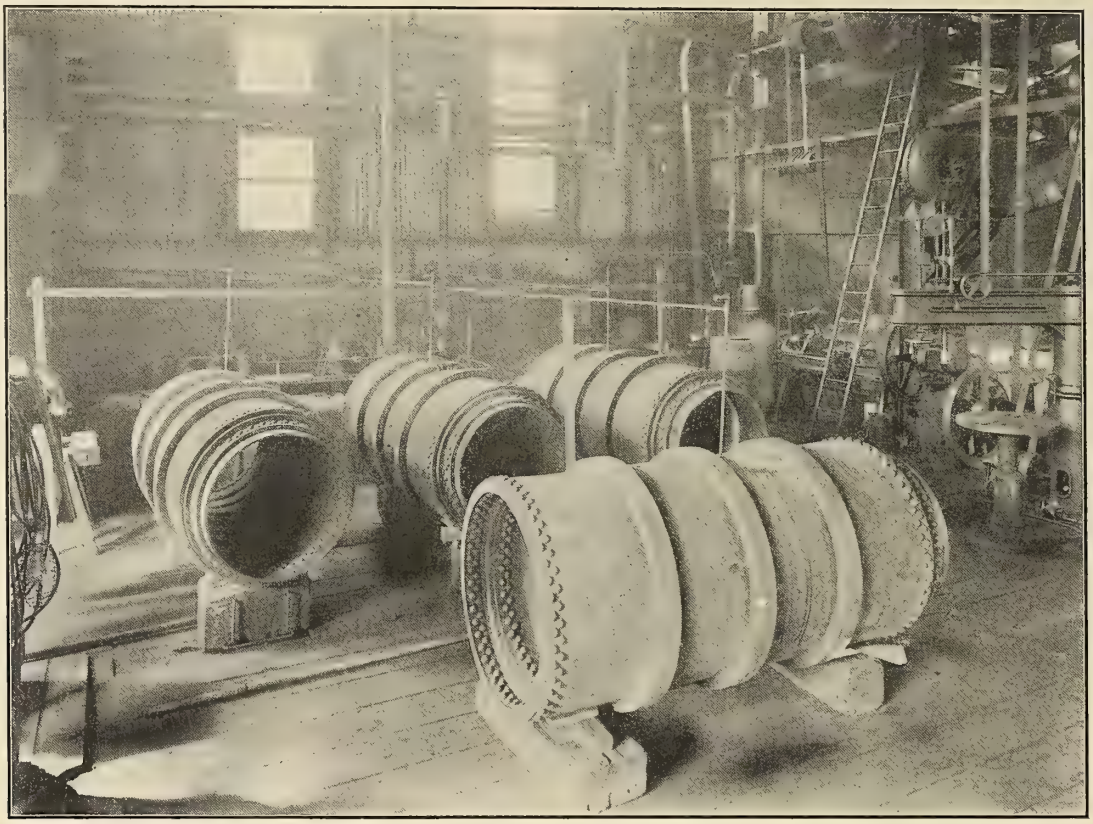
In all cases of accident at sea do not get confused, above all things, for quick wit in emergency will get one out of a very tight place in the majority of cases. It is often the brag of a good gas-engine runner that he has never been towed home, but got back under his own power, even if it took him all night to make the necessary repairs.

(The End.)

Steamship Cedric.—The new White Star Line steamship *Cedric*, which is practically the same as the *Celtic*, arrived in New York on her maiden voyage on February 21. The general dimensions of this ship are: Length over all, 700 feet; breadth, extreme, 75 feet; depth, 49 feet 4 inches. A description of the ship will be given to readers of Marine Engineering in the April number.

New Canadian Shipyard.—It is reported that the Canadian Pacific Railway will build several large freight steamers for service on the Great Lakes, and, with this end in view, the road will, it is reported, give financial backing to the extent of $1,000,000 to the Bertrams, of Toronto, for building and equipping a plant where ships may be completely equipped. The site for the plant is stated to be Sandwich, Ontario, opposite Detroit, Mich.

Lumber Raft.—On the Pacific Coast great quantities of lumber are brought down the rivers and made up into gigantic and immense rafts and then towed down the coast to San Francisco or other ports where lumber is to be delivered. One raft, containing 8,000,000 feet of piling, recently built by the Robertson Raft Company, was towed from the Columbia river to the Golden Gate by two sea-going tugs in eight days.



BROWN FURNACES MOUNTED FOR TESTING IN THE MACHINE SHOP OF THE CRAMP DRY-DOCK COMPANY.

TESTS OF CORRUGATED FURNACES.

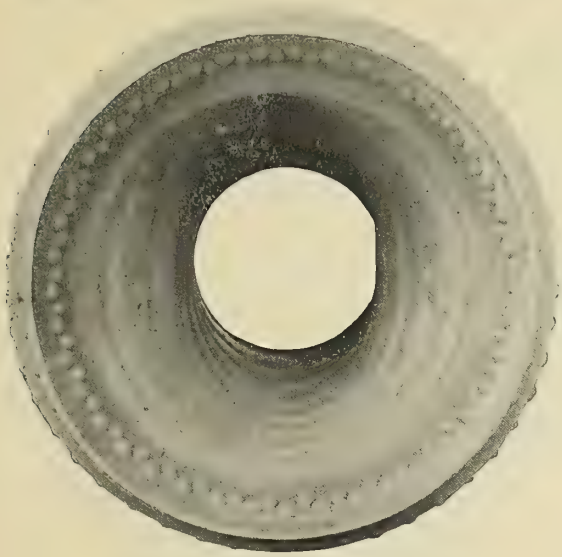
A most interesting series of tests was made at the works of the Cramp Dry-Dock Company, Philadelphia, Pa., on February 3, 1903, of the Brown patent improved corrugated furnace. The tests were held under the direction of the Board of Supervising Inspectors of Steam Vessels and were witnessed by a representative gathering of about a hundred engineers from the shipyards and boiler works throughout the country. The furnace is the latest design of John Brown and Company, Sheffield, whose furnaces have for many years been in general use on the other side of the Atlantic in cylindrical boilers, both on land and sea; the selling agents in this country are Worth, Baily and Company, Philadelphia, Pa.

The Brown furnace differs from other furnaces on the market in, first, its method of manufacture, and, second, its design. The plate selected is of greater thickness than that of the finished furnace. It is then heated and passed through the corrugating rolls. Here the corrugations are left of heavier metal than the rest of the plate, for, when the furnace is in place, under pressure in the boiler these corrugations, being at the greatest diameter, are subjected to the greatest stress. The corrugations are spaced nine inches center to center, and the space between them is left flat. This surface is left flat because it is found to transmit more heat than a corrugated surface, and also because it is an easier furnace to fire.

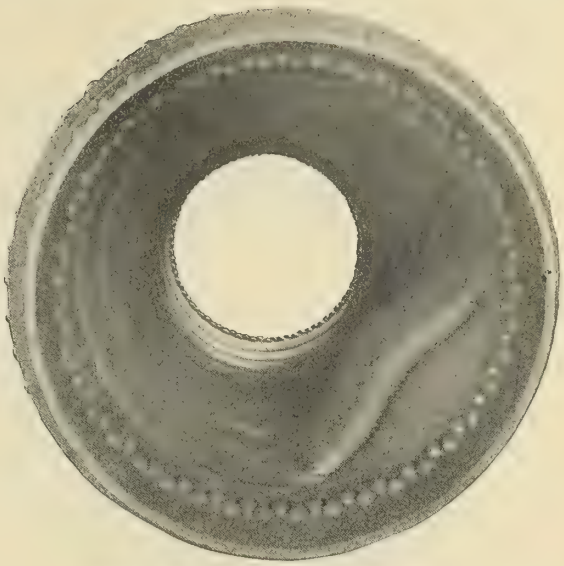
The plate is sheared cold, and the welding edges bevel sheared. It is then wrapped into the form of a cylinder, the edges to be welded brought to a welding heat and welded by special hammers. The weld thus being made after rolling the corrugations, it is not subjected to the strains incident to passing through the rolls. The furnace is then placed vertically in an annealing furnace, and, when at the proper heat, is brought to a special hydraulic press which takes out the flat produced by the welding operation. It is then pickled and dressed, shaped, flanged, cut to size, adjusted to iron templates, and once more annealed. As it has been rolled from a thicker plate, the ends may be left heavier for riveting to front or back connections.

The steel used is of medium tensile strength of about 60,000 pounds, as experience has shown that plates in contact with flame should be of comparative low tensile strength and correspondingly greater ductility.

The furnaces tested were four in number, and mounted as shown on drawing, page 145, and in the engraving above. Each was securely riveted within a heavy cylinder braced by 4 by 2-inch bands. The space between cylinder and furnace was connected up to a pressure pump and accumulator, as will be seen in the engraving, by one-quarter-inch piping, and a hydrostatic pressure applied till the furnace collapsed. Horizontal and vertical diametrical measurements were taken by gage on the first, fourth, and seventh flats, first before the pressure was applied, and then at increasing pressures up to 700 to 800 pounds per square inch. The pressure was at that point released and the dimensions taken to see if there had been any permanent set. After this the pressure was applied until rupture occurred.



FURNACE NO. 103 AFTER DESTRUCTIVE TEST.



FURNACE NO. 104 AFTER DESTRUCTIVE TEST.

1st. 4th. 7th Flats.

Pressure in lbs. per Square Inch	Front.		Middle.		Back.		Remarks.
	Vert.	Hor.	Vert.	Hor.	Vert.	Hor.	
Original Diameter.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	
0	$29\frac{5}{32}$	$29\frac{4}{32}$	$28\frac{31}{32}$	$28\frac{31}{32}$	$28\frac{29}{32}$	$28\frac{29}{32}$	An entire collapse end to end at top and bottom.
400	$29\frac{9}{64}$	$29\frac{7}{64}$	" f.	"	$28\frac{57}{64}$	$28\frac{57}{64}$	
600	$29\frac{4}{32}$	$29\frac{7}{64}$	$28\frac{81}{64}$	"	$28\frac{28}{32}$	"	
700	"	$29\frac{3}{32}$	"	"	"	$28\frac{28}{32}$	Scale cracked at 1,050 lbs. Average thickness .407.
0	$29\frac{9}{64}$	$29\frac{4}{32}$	$28\frac{31}{32}$	$28\frac{81}{64}$	$28\frac{29}{32}$	$28\frac{29}{32}$	Constant 15,356.
800	$29\frac{4}{32}$	$29\frac{3}{32}$	$28\frac{29}{32}$	$28\frac{29}{32}$	$28\frac{28}{32}$	$28\frac{57}{64}$	
950	Pump valved choked; pressure released.						
0	$29\frac{9}{64}$	$29\frac{4}{32}$	$28\frac{31}{32}$	$28\frac{31}{32}$	$28\frac{29}{32}$	$28\frac{59}{64}$	
1250	$27\frac{30}{32}$	$30\frac{17}{64}$	$22\frac{3}{16}$	$34\frac{12}{32}$	$25\frac{26}{32}$	$30\frac{16}{32}$	After collapse.

FURNACE NO. 103.

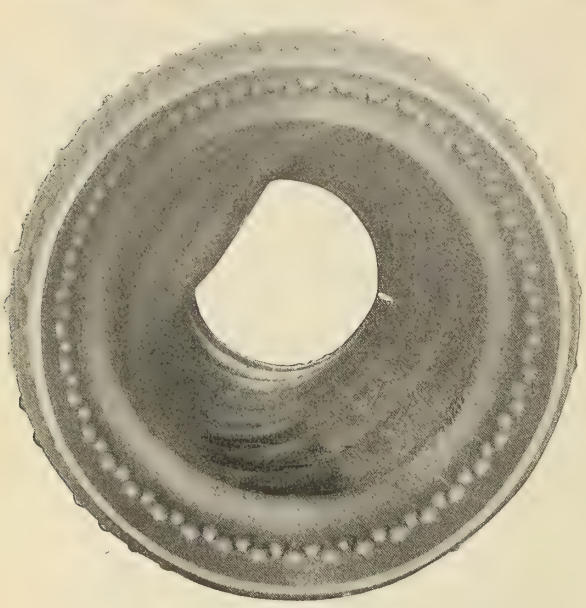
	Tensile strength.	Elongation.	Reduction area.
Original plate	61,944 pounds.	26 per cent. in 8 inches.	
Sample from furnace	59,029 "	22.9 " " 6 " (broke at neck).	64.4 per cent.

1st. 4th. 7th Flats.

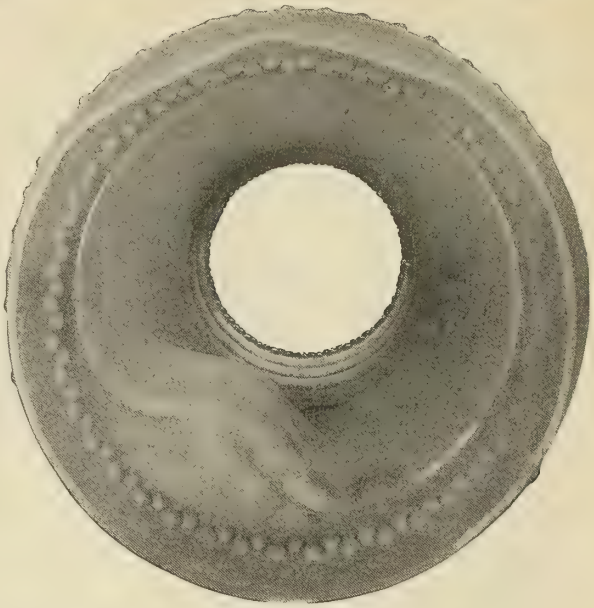
Pressure in lbs. per Square Inch.	Front.		Middle.		Back.		Remarks.
	Vert.	Hor.	Vert.	Hor.	Vert.	Hor.	
Original Diameter	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	
0	$32\frac{3}{32}$	$32\frac{4}{32}$	$31\frac{28}{32}$	$31\frac{26}{32}$	$31\frac{31}{32}$	$31\frac{28}{32}$	As collapse was deepening, second rip burst at 800 lbs. in circumference fracture 9 1-2 ins. long across weld. Fracture crystaline.
500	$32\frac{2}{32}$	$32\frac{2}{32}$	"	$31\frac{25}{32}$	$31\frac{30}{32}$	$31\frac{27}{32}$	
725	No dimensions taken.						
0	$32\frac{3}{32}$	$32\frac{4}{32}$	$31\frac{28}{32}$	$31\frac{26}{32}$	$31\frac{30}{32}$	$31\frac{27}{32}$	Scale cracked at 1,200 lbs.
1200	Gas plug blew out, releasing pressure suddenly.						Mean thickness U. S. .478.
1300			$32\frac{2}{32}$	$31\frac{30}{32}$	$31\frac{18}{32}$	$31\frac{26}{32}$	Constant 14,958.

FURNACE NO. 104.

	Tensile strength.	Elongation.	Reduction area.
Original plate	62,392 pounds.	22 per cent. in 8 inches.	
Sample from furnace	59,829 "	30.2 " " 6 "	64.1 per cent.



NO. 105.



NO. 106.

FURNACES AFTER TESTING TO DESTRUCTION.

1st. 4th. 7th Flats.

Pressure in lbs. per Square Inch.	Front.		Middle.		Back.		Remarks.
	Vert.	Hor.	Vert.	Hor.	Vert.	Hor.	
Original Diameter.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	
0	$29\frac{3}{32}$	$29\frac{3}{32}$	29	$28\frac{31}{32}$	$29\frac{6}{32}$	$29\frac{7}{32}$	Scale cracked at 1,110. Mean thickness U. S. .460.
500	"	"	$28\frac{31}{32}$	"	$29\frac{11}{64}$	$29\frac{6}{32}$	
700	No dimensions taken.						Constant 13,586.
0	$29\frac{3}{32}$	$29\frac{3}{32}$	$28\frac{63}{64}$	$28\frac{31}{32}$	$29\frac{6}{32}$	$29\frac{7}{32}$	
1250	$29\frac{19}{64}$	$28\frac{1}{2}$			$29\frac{13}{32}$	$28\frac{31}{32}$	After collapse.

FURNACE NO. 105.

	Tensile strength.	Elongation.	Reduction Area.
Original plate	60,704 pounds,	29 per cent. in 8 inches.	
Sample from furnace	56,909 "	27 " " 6 "	63 4 per cent.

1st. 4th. 7th Flats.

Pressure in lbs. per Square Inch.	Front.		Middle.		Back.		Remarks.
	Vert.	Hor.	Vert.	Hor.	Vert.	Hor.	
Original Diameter.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	
0	$32\frac{23}{64}$	$32\frac{16}{64}$	$32\frac{2}{64}$	$32\frac{4}{64}$	$32\frac{10}{64}$	$32\frac{11}{64}$	Average thickness .384 ins. Scale commenced to crack 850 lbs.
400	$32\frac{22}{64}$	$32\frac{16}{64}$	32	$32\frac{2}{64}$	$32\frac{10}{64}$	$32\frac{10}{64}$	
600	"	"	"	"	"	"	Constant 13,778.
0	$32\frac{22}{64}$	$32\frac{16}{64}$	$32\frac{2}{64}$	$32\frac{4}{64}$	$32\frac{10}{64}$	$32\frac{12}{64}$	
962	$24\frac{18}{64}$	$32\frac{56}{64}$	$31\frac{56}{64}$	$32\frac{6}{32}$	$32\frac{4}{32}$	$32\frac{4}{32}$	After collapse.

FURNACE NO. 106.

	Tensile strength.	Elongation.	Reduction Area.
Original plate	62,168 pounds.	31 per cent. in 8 inches.	
Sample from furnace	60,180 "	30.2 " " 6 "	63 4 per cent.

Points for taking the dimensions were marked on the plate by a center punch. In the preceding tables will be found the records of each furnace. The tests were eminently successful, and, by the way, are the only ones of the kind of which there is any record held in this country. The Board of Supervisors allowed the maker to use the same rule for determining the thickness and diameter for a given steam pressure as is laid down by the British Board of Trade, except that they required a factor of safety of 6 instead of 5 1-4 as is practiced in England. The formulæ are as follows:

Let C = constant.
T = thickness of plate in inches.
D = smallest outside diameter in inches.
P = working pressure, pounds per square inch.
P' = collapsing pressure, " " " "

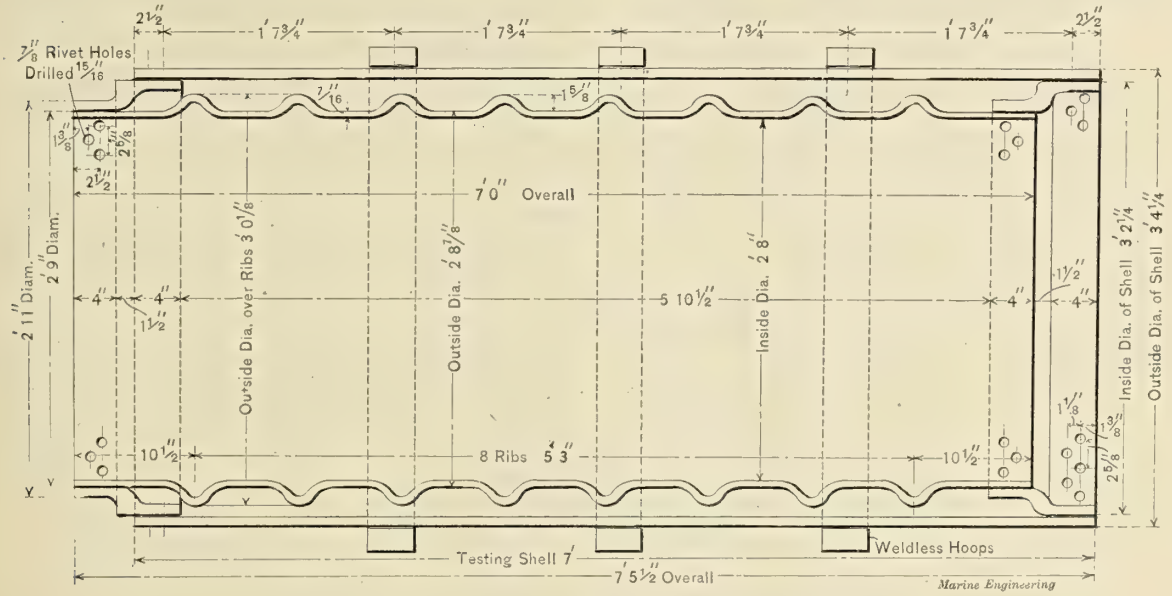
Then $C = \frac{P'D}{6T}$ and $P = \frac{CT}{D}$

Propellers for British Cruisers.

In connection with the four first-class cruisers *Good Hope, King Alfred, Drake,* and *Leviathan,* a very interesting series of trials has been conducted, according to the *London Engineer.* These ships, of 500 feet length, 14,100 tons displacement, with Belleville boilers of about 72,000 square feet of heating surface and 2,310 square feet of grate surface, gave on full power trials the following results:

	I. H. P.	Speed knots.	Coal per hour per I. H. P.
Good Hope	31,119	23.05 m.m.	1.91
King Alfred	30,950	23.65 log	1.81
Drake	30,864	23.05 "	1.82
Leviathan	31,208	23.23 m.m.	1.94

The trials were first conducted with the *Drake* to determine the question of propeller efficiency. The loss in slip increased at a very rapid rate after 22 knots had been attained, and it was decided to change the original propellers, which had been proportioned as a result of tests made at the government experiment tank. The new propellers, first fitted to the *Drake,* had a greater ratio of surface to diameter, with the result that at full power the speed became 24 knots, or 1 knot in excess of the contract trial. Since then similarly-proportioned propellers have been ordered for all of the ships, so that each ship may have a speed of 1 knot over that recorded in this table. Corresponding propeller trials have also been carried out and are still in progress in connection with the vessels of the *County* class.


7/8" Rivet Holes Drilled 15/16"
2 11" Diam.
2 9" Diam.
7 0" Overall
Outside Dia. over Ribs 3 0 1/8
Outside Dia. 2 8 3/8
Inside Dia. 2 8"
Inside Dia. of Shell 3 2 1/4
Outside Dia. of Shell 3 4 1/4
8 Ribs 5 3"
5 10 1/2"
Testing Shell 7'
7 5 1/2" Overall
Weldless Hoops
Marine Engineering

BROWN PATENT CORRUGATED FURNACE MOUNTED IN TESTING CYLINDER.

Views of the furnaces after the tests are given on the preceding pages. No. 106 showed a most unique collapse. The soft spot at the middle in the front end of the furnace was bulged out and the metal flowed cold, so that the corrugations came out in a flat surface. On No. 104, at a pressure of 1,200 pounds, a plug blew out in the side of the cylinder, relieving the pressure. The material of the furnace had evidently passed the elastic limit, because, when the pressure was again applied and 1,300 pounds reached, a fracture was discovered in one of the ribs, which burst before the entire pressure could be relieved. The other tests need no comment. The constant allowed by the Board is 14,000.

The entertainment offered to the guests was as hospitable as the tests were successful. The guests were taken from the shipyard to the Manufacturers' Club, where a fine banquet was provided under the auspices of Messrs. Worth, Baily and Company, General Dumont and the Supervising Board of Inspectors being the guests of honor. Many of those present were called upon for speeches by Mr. W. Wallace, marine engineer of John Brown and Company.

Racing Motor Boats.—Added interest will doubtless be taken in motor boats, due to the action of the New York yachtsmen forming the National Boat Association. Up to the present this type of racing has not been indulged in to much extent in this country, but now, under the encouragement of the National Association, interest in these races will be greatly increased.

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Notice to Advertisers.

Changes to be made in copy, or in orders for advertisements, must be in our hands not later than the 15th of the month, to insure the carrying out of such instructions in the issue of the month following.

EXCEPT in the Great Lakes yards, the outlook for new work in shipbuilding for the next year is not especially encouraging. In the Great Lakes district the yards have practically all the work they can take care of, and in some cases new work has been of necessity declined. Orders for this new tonnage have, of course, been predicated on a confidence in the continuance of the present commercial conditions during the next few years, and in the capacity of the modern ship to more than hold its own in competition with the less efficient ship of a decade or more ago. This, again, will depend on the general condition of business and industry as governed by the complex relations of demand and supply, inherent business confidence, crops and natural products, relations of labor and capital, etc., etc. Whatever the next few years may hold for the business of the country, the Great Lakes operators, by their recent orders, have signified their faith in a continuance, for some few years at least, of the present prosperous conditions.

With the yards on the Atlantic coast the condition is by no means as satisfactory. None of the yards can be considered full, and all are doubtless looking for more work. The conditions here are so different from those on the Lakes that no comparison between the two can be readily made without extended discussion. While the Coast builders would all be most gratified with such general conditions as have recently prevailed on the Lakes, and while all who are interested in maritime affairs will agree that such a condition is much to be desired, the problem remains—how to bring it about?

Is the condition one which calls for new legislation, for a readjustment of business methods, for an increase of confidence in ships and shipping as a field for investment, for a readjustment of prices for ships, or of the relations between the costs of labor, material, and administration, or for all these combined and with others perhaps unnamed? This is a question hardly to be answered in a few lines on the editorial pages. It is one where experts disagree, and where, after all, it comes largely to a matter of opinion. We have asked the question as one worthy of the most careful and exhaustive study on the part of all who are interested in maritime matters, and will only add the further thought that any study of this question which takes for the possible field of operation of our deep-water shipping anything less than the wide world of commerce will be only partial and unsatisfactory in its conclusions. We must take our place with other commercial nations, our solution of the problem must be influenced by the solutions which other nations are finding, and, in general, we must study the situation as a part of the world problem of commerce and commercial exchange.

WE would call the attention of our readers to the article published on another page and relating to the powering and propulsion of small boats. The problem of propeller design in particular is so little understood by the great body of amateurs who are interested in motor boats that we believe the relatively simple methods given will serve effectively to aid those who wish to give some study to this part of the problem. Quite recently a correspondent writes to learn if the speed at which the boat is going will have any great influence on the problem of design, or on the necessary size of propeller for a given power. The form of the equations given shows the close and vitally important relation between the size and the speed for a given power. It shows clearly that, with a given power, the higher the speed the smaller the propeller, and *vice versa*. Propeller

design can never be reduced to the mere mechanical solution of a formula or series of formulæ. It must necessarily contain a very considerable element of judgment based on experience, and, at best, it is a problem intricate in character and in many details calling for more experimental work before the relations of various features are fully understood. For the purposes of those interested in small boats, however, relatively simpler means may be made to give satisfactory results, and to this end the article referred to has been prepared.

THE bill recently introduced into the Senate by Senator Frye and into the House by Representative Dalzell, looking to the appointment of a commission to suggest an entire revision of the code of rules and regulations relating to the construction and inspection of marine boilers, is one of the most important steps taken in recent years in connection with maritime legislation. The existing laws and the rules which are their interpretation have grown up piecemeal and without natural order or sequence. In many features contradictions and inconsistencies appear, and the correct interpretation is not always readily found. In many respects also the rules have fallen behind the advance of engineering progress, and revision and addition are needed in order to bring them into closer harmony with the best practice of the day, and with the possibilities of present-day methods and materials. The officers of the Board of Supervising Inspectors have long realized many of these facts, but have been hampered by the existing laws. We trust that the bill may speedily become a law and that the Commission will lose no time in setting itself to work on the task before it. The text of the bill provides that the Commission shall submit to the Secretary of the Treasury a preliminary report on or before November 1 next, that such report shall be freely furnished to all who may desire a copy, and that the Commission shall give an opportunity to all interested parties to submit any protest, suggestion, or recommendation in regard to changes or amendments that should, in their opinion, be made. After the careful consideration of such matter as may be thus presented, the Commission shall make a final report on or before February 1, 1904, for transmission to Congress. This is a promising programme, and we shall look with confidence for a revision of the laws relating to boiler design, construction, and inspection, which shall place the United States abreast of the best engineering practice and possibilities of the day.

WE should like to give a word of advice to our colleagues of the daily press; not that they will perhaps heed it, but that will not be our fault. There is hardly a day passes but that the daily press somewhere heralds some world-startling or epoch-making discovery in engineering or science. Ships in the form of some unthinkable monstrosity or propelled by some equally absurd propelling device are to cross the ocean in three days or less; some wonderful form of electric generator has been devised which is immediately to revolutionize the entire art of electrical engineering; some form of motor has been invented which, when sifted down to the ultimate analysis, means perpetual motion, or a new form of Keely motor; and so on *ad mirabilem*. In these particulars the situation has come to be such that no one in the least acquainted with engineering or science gives the slightest credence to any statements of this character which he may see in the daily press. Those who do not know may give a momentary credence, but only to find that they were following a blind teacher. It is not too much to say that in such matters the daily press is simply the laughing stock of the scientific and engineering fraternity; if such articles are read at all, it is simply as a joke.

As to how such "science fakes" are to be avoided, we can hardly undertake to say in all cases. An engineer on the staff of a large daily would often be of the greatest use in connection with such matters, and in the correct reporting of matters with an engineering bearing. The illustration of a marine boiler according to the ideas of the artist of a daily paper, and which we reproduced in our February number, is an illustration in point. Another way would be to consign to the waste basket all such reports as seem to bear on the marvellous in regard to engineering or science, on the principle that it may be better to fail to notice some meritorious discovery rather than to mislead, ninety-nine times in a hundred, the unknowing public and fill up space with absurd and impossible nonsense. How such absurdities originate would make an interesting study, but with that phase of the question we are not here concerned. We trust that our friends of the daily press will accept our reference to this matter in the kindly way in which it is intended, and we shall live in the hope that on the larger dailies the office of engineering and scientific expert may yet become one of established and recognized importance.

THE SOLILOQUY OF A 3,000 H. P. TRIPLE.

BY C. A. MCALLISTER.

"Well, here I go again," mused the main engine of a coasting steamer one cold winter afternoon, as the ship passed out by Sandy Hook on one of her weekly trips to the southward. "I have been at this business steadily now for over six years, and I am getting a little bit tired of this eternal grind for a living. Here I have kept at it continually without a breakdown, and haven't had to stop once or ever caused delay by not being ready on time. True, on several occasions I have grown hot in the crank pins or in my guides, but that was because I was out of sorts on account of bad treatment. Well do I remember when they took me out on my first trial trip down the Delaware river. It was a bright morning in June and I was feeling like a colt, with my three thousand horses ready for the fray. My, how many complimentary things were said about me that day, especially when I kicked up my heels and made a spurt of 150 turns for an hour or so! Oh! the attention they gave me then! Those grooms were so anxious about my welfare that they were around me in swarms—felt of me all over, patted me, and, oh! such fine large doses of delicious oil as they did pour around my working parts; yes, and gave me a drink of good mellow old cylinder oil about every ten minutes. Then there were a lot of wise-looking youngsters standing around my cylinders, and they would take what they called indicator cards off three little nickel-plated instruments they had hitched up to me. Another youngster, who wore glasses and looked as if he would break in two if a puff of steam would strike him, had a big book in his hand and was putting down all kinds of figures from my gages. After a set of cards was taken the little boys would get together, look wise, make a lot of figures, and then the leader of them would whisper something to the superintendent of the shipyard, whereat that mogul would give a self-satisfied smile and announce to old man Jones, the president of our company, that I was doing a little better than 3,300. That, of course, tickled the old man, but I could feel in my frame that I wasn't doing a bit more than 3,000. Those draftsmen they send out on trial trips are regular wizards for jockeying figures to make them come out as they want them. When I got back to the wharf after the trial trip I was probably the 'chestiest' triple that ever came out of a shop. You see I was young and gullible, and such remarks as, 'Why, I never saw such a smooth-running engine'; 'Absolutely no vibrations'; 'We didn't have to use a drop of water on her, anywhere'; 'Never slowed down from start to finish'; 'The power is more evenly distributed among the three cylinders than I ever saw before'; 'You can throw her over with one finger on the reversing lever'; 'Stuffing boxes absolutely tight,' etc., nearly turned my cylinder heads. Since I have grown somewhat more accustomed to the ways of this hard, cruel world, I realize that any old scrap pile of an engine can get such bouquets as those thrown at them, providing the builders supply enough champagne to the guests.

"Alas and alack! but times have changed since that June morning. I no longer have such compliments thrown at me these days. Just think if it, that new red-headed First Assistant had the nerve to call me an old 'coffee mill' this afternoon when I was warming up at the wharf. I don't like that fellow anyhow; he has been chief of one of these millionaires' yachts all summer and his head is a little swollen. Just because there are not many polished brass bands around me, and no gallery with plate-glass windows where the passengers can come in and look me over, he thinks that I am a cheap skate. I don't believe he has ever stood a midwatch in a gale of wind in his whole life. He has a whole lot of things to learn yet. I wish they had promoted that young Second Assistant instead of running in that new fellow. The youngster is all right and tends to his business right up to the handle. He has only been with us two years now, but I feel easier when he is on watch than with any of the others. Of course, the old Chief is all right, but he doesn't take a watch now that we have a Third Assistant. The Chief and I have been together now ever since I first came out and we have never had a row yet. Here lately, though, the old gentleman is getting awfully stingy with me.

"There are many repairs I ought to have made in my pins and joints. The fact of the matter is, I need a vacation. I'm getting awfully tired of this everlasting thump in my after main bearing. They say I'm a little out of line, and that my bottom brass is worn down a trifle too much. Well, if they do not overhaul me a little and give me some rest and treatment I'll surprise them some day and break down out at sea. Then the owners will have something to kick about, if they have a nice large salvage bill to pay. Another thing I don't like is that they are getting too mean with their cylinder oil. I heard the Chief tell the young fellows to-day that there was too much oil on the crown sheets of the boilers the last trip and that they must cut down on my allowance. I don't see why they have to cut off my rations just to accommodate those boilers. I am sure the boilers never give me any consideration. Here lately every chance they get they cough up a lot of water and shoot it into me through my main steam pipe. They nearly choked me one night on the last trip up the coast. I wheezed and strangled so much that I thought for a time my heads would burst. What that little Second Assistant said to the water tender through the speaking tube was almost enough to melt the tube. That boy always looks after my interests and takes my side of the case.

"The boilers are getting lazy of late and don't furnish me as much steam as they used to. Then the engineeers try to take out their spite on me by jockeying around with the throttle and my cut-offs. If they would get in and read the riot act to those four boilers they might accomplish something. I'm getting a little old, but I'll give them the revolutions yet if they only keep the gas coming. Every once in a while I can hear the boilers grumbling among themselves about their tubes being stopped up, grease and dirt on their heating surfaces, poor coal in their furnaces, firemen that don't know their business, oily feed water, and a hundred-and-one other complaints they have. Boilers are very touchy things anyhow and growl at the slightest provocation. Glad I'm not a boiler; they're entirely too cranky!"

"Huh!" chimed in the Independent Air Pump, who had been listening to the reverie of the Triple, "you needn't talk about crankiness; any one who has three cranks all going around at once has no kick coming to him." The Air Pump was one of these direct-acting fellows, and never lost an opportunity to guy the Triple about his cranks.

"Well, it's not becoming of you to criticise my cranks," retorted the Triple; "if you only had a couple of them yourself there would be no chance for you sneaking in your little half strokes as you have been doing recently, and knocking my vacuum down so that at times I think I am breathing out into the open air."

"Cranks wouldn't help your old vacuum any," said the Air Pump; "all you need to do is to wash out your condenser with a little soda and lye water once in a while. It's surprising how dirty some people can be; why, it's a wonder the quarantine officers don't hold you up some time."

"That will hold him for a while," chimed in the Main Feed Pump, who was not particularly friendly to the Triple. The boss of the engine room was plainly nettled by such persiflage, and as the screw propeller lifted out of the water considerably at that instant, he gave vent to his wrath by buzzing around at the rate of about five hundred revolutions a minute, which so startled the new First Assistant, who was on watch at the time, that he nearly dropped the pipe out of his mouth.

After he had been throttled down the old Triple could hardly contain his wrath at the impudence of what he considered to be such insignificant members of the engine-room family. Finally he launched forth in the following tirade:

"As for you, Mr. Feed Pump, you are simply so small and unimportant that any remarks you may make don't attract attention from any one. A pump which everybody places so little reliance upon that every ship has to carry several other pumps to do its work whenever it breaks down, and that happens pretty often, need not expect to be noticed by me. It's well they call you a 'duplex,' for I don't know of any one who practices more duplicity than you.

"As to your remarks about my dirty condenser, Mr. Independent Air Pump, I must say that they come in very poor grace. Of all the slovenly, dirty-looking objects around this engine room you are the limit. Look at you leaking water around your pump rods and spilling it all over the floor plates; look at that dirty streak of cylinder oil sizzling on your steam chest. You are getting entirely too frivolous lately and you don't half attend to your work, although every one remarks that you use up steam enough to run a dozen pumps; and it is common talk around the engine room that you have a regular cylinder-oil jag every day. Here lately they are giving you as much oil to drink as I have myself, and any one knows that I can stand it better than you. It was a sorry day for you Air Pumps when you had your apron strings, or links as we call them, cut from us main engines. You have become entirely too 'independent' for your own good. When we had you hitched to one of our crossheads you simply had to attend to business whether you wanted to or no, and it was precious little attention you ever got from any of these engineers or oilers. I wish we could get along without you entirely; and I really believe I could, as I have a connection from my condenser over to that little Auxiliary Feed Pump. He could pump the water out of my system and prevent me from having the dropsy. Why don't you behave yourself like that quiet little Contrifugal Circulator over there back of you? He goes at his business day in and day out and never gets gay like you do. If it wasn't for that faithful young fellow I believe I'd burn up when we strike the warm water south of Hatteras. He never complains about running up twice as fast as he is going now, whenever the sea water gets warm."

The Air Pump was getting vexed under this curtain lecture from old Triple, and, although he made no reply, every one noticed that he was getting sulky. Finally, to accentuate his feelings, he made three or four quick strokes and stopped dead still. Old Triple did not notice it for a moment or so, but finally he too had to slow down, so much so that he began to groan and his low-pressure valve ratted frightfully. This scared the new First Assistant nearly out of a year's growth, and the way he and the two oilers scampered around was a sight to behold. One of the oilers said that the Air Pump was simply balky and rushed up and struck it a sharp blow on the end of the valve chest. With a sudden groan the sulky pump started right up again, the needle on the vacuum gage went around to 25 inches, and old Triple resumed his former gait.

"I thought you could get along without me," tauntingly remarked the Air Pump. "So I can," replied old Triple, "but they wouldn't connect up the Auxiliary Feed. I'm glad they gave you that rap with the sledge, and hope you get a worse one the next time you attempt any of your dirty tricks." The First Assistant reported to the Chief the next day that one of the air pump valves had become dry on his watch and that the pump had stopped. Little did he know of the bad feeling existing between the main engine and the pump.

Very little more was said in the engine room that night. Old Triple had the sulks and kept up a continuous pound and thump all night. At four o'clock when his friend the Second Assistant came on watch, he was given a half pint of cylinder oil, which made the old boy feel so good all over that he stopped his pounding. The Second Assistant had directed the oiler to pump a little of the oil into the first receiver, and old Triple remarked: "There, that goes right to the spot," as he had a pain around his intermediate valve. As he fell to ruminating again he wondered why he could not have all the cylinder oil he wanted. "I know it is said to be a bad habit, but I don't see that it hurts me very much. I suppose that after a while they will have me signing a pledge to abstain from oil altogether, as these men do when they want to stop drinking liquor. If they had not given me the first drink of this oil I wouldn't have to break myself away from the habit now."

Just then the little shrill voice of the Main Speed Pump was heard: "Look out, you old toper, or the first thing you know they will be sending you to a Keeley Institute." "Well, I have been getting samples of the Black Lead (graphite) cure, so I presume I

could stand the gold cure, but even that would be preferable to being compelled to work in the same room with such an insignificant shrimp as you," came back old Triple. "I'd rather be a shrimp than an old oil-soaked whale," chuckled the Feed Pump, and this remark raised a metallic laugh among the denizens of the engine room, at the expense of main engine, who restrained his rage, but became somewhat heated by taking an extra turn or so.

Hour after hour the time passed, and hardly a word was spoken; the Air Pump kept up a dignified silence; the Feed Pump would occasionally shout out something to try to amuse his companions, but Old Triple would pay no attention to him. The old fellow looked weary and was plainly not at his best, as his revolutions were not averaging much over 120. Occasionally he would break out into a rhythmic chant of "I-need-a-rest, I-need-a-rest, I-need-a-rest," and keep it up for an hour at a time. The impish Feed Pump would, on such occasions, arouse his ire by chanting in his quick, jerky voice: "O-give-us-a-rest, O-give-us-a-rest, O-give-us-a-rest," with the accent on the "us."

The Chief Engineer, who had been with Old Triple long enough to interpret some of his talk, was quite worried whenever he heard the main engine fall into these despondent fits, and the "I-need-a-rest" chant rather haunted him. Accordingly when the steamer arrived at her last stopping place on the outbound trip, he and the assistants gave Old Triple a thorough looking over, making many notes as to the condition of his various parts. These were embodied in a report to the Superintending Engineer, and forwarded by mail

On the homeward-bound trip Old Triple resumed his chanting, and as they were going out of port he would repeat: "It's-the-same-old-grind, it's-the-same-old-grind, it's-the-same-old-grind." One night when the First Assistant was on watch, the engine got so mad at things in general that he got hot in the after main bearing. Although they poured oil and water on it he refused to let it cool off until the Second Assistant came on watch. "I like to have fun with that red-headed chap who called me an 'old coffee mill,'" the Triple remarked, and then cooled down and ran as nicely as ever. He and the Air Pump were no longer on speaking terms, and each wore an air of haughty indifference to the other during the entire trip northward. The impudent Main Feed had broken down entirely the second night out, and the Auxiliary had taken his place. As he was a quiet, unassuming chap who attended strictly to his own business, there was very little gossip going on.

Old Triple mused as they were going up the bay that he would be wiped down when they got to the wharf, have some adjustments made to his crank pins, one of his top brasses scraped, and then in three days off he would start again on another monotonous trip. Much to his surprise, just after the jingle bell was sounded, the Chief announced that the ship was to be hauled off and given a month in which to make repairs at a machine shop. Old Triple's chanting and the Chief's report had had the desired effect.

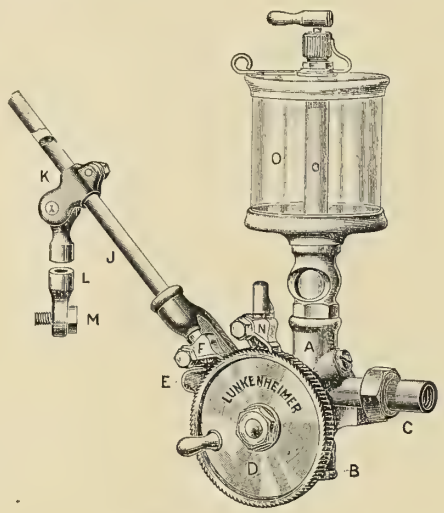
Moral: *All ye good seafaring engineers, learn to know your engines and try to understand their talk.*

ENGINEERING SPECIALTIES.

Mechanical Oil Cup.

A mechanical system of oil feed is generally considered necessary on all large engines, and a method of lubrication which was found, after experimenting, to be reliable and positive, is shown in the devices here illustrated, made by the Lunkenheimer Company, Cincinnati, Ohio. Motion may be taken for driving the mechanism from an eccentric rod or other moving part of the engine by means of the rod *J* and couplings *K* and *M*. The rod *J* operates the ratchet wheel *D* through the clutches *F* and *N*, and the motion is transmitted to the piston *E* by the crank-pin mechanism.

The ratchet wheel *D* is provided with a handle whereby it can be rotated by hand in case it is desirable to force a quantity of oil at any time, as, for example, when starting the engine. By moving the part *K* up

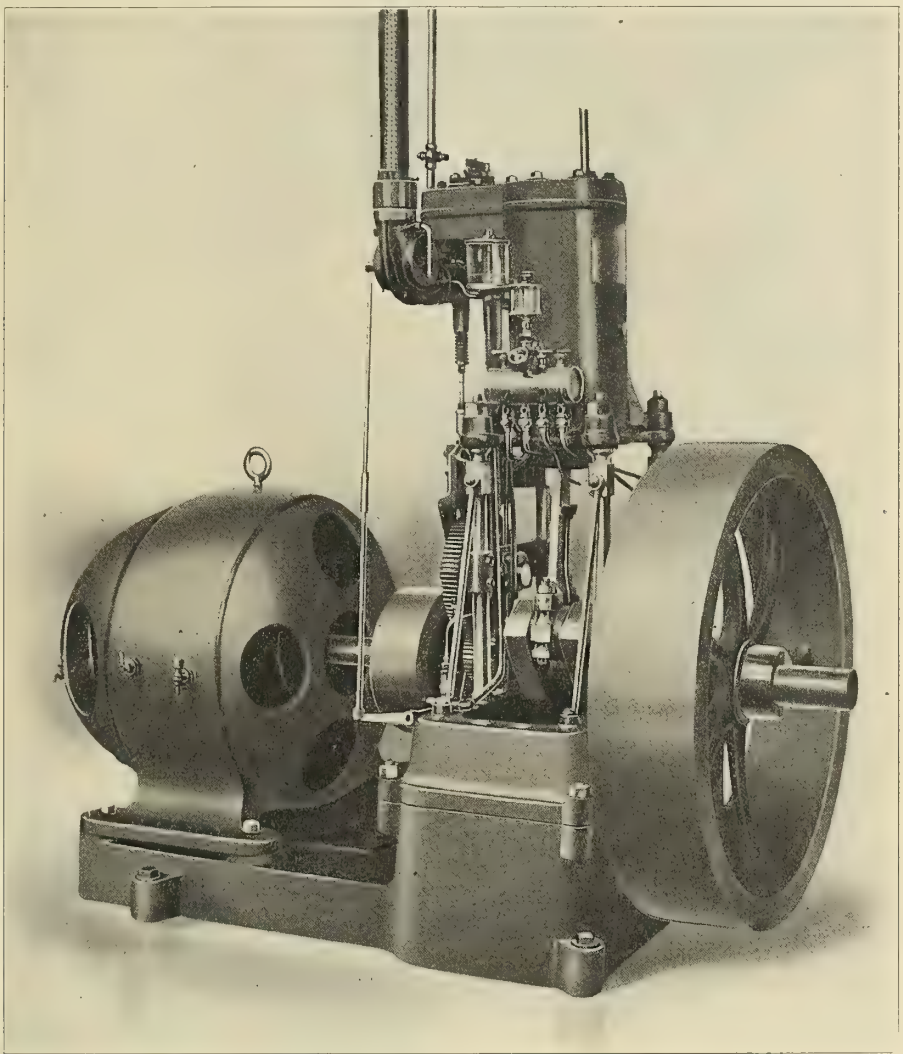


OIL FEED PUMP.

or down the rod, the stroke of the pump can be lengthened or shortened, as desired, thus regulating the amount of oil fed by the pump independent of the feed from the oil cup. The joints of the cup are tight, the sight-feed glass being packed so as to prevent the access of air that would have a tendency to cause the cup to feed after the engine had ceased running. This construction and the use of check valves in the pump prevent oil supply from flooding. The outlet *C* is piped to the steam pipe or chest of the engine, and the spring check valve should be placed as near the end of the pipe as possible, preferably into the steam pipe. The bottom of the pump body, *B*, is tapped with a 1-2-inch pipe thread to receive a stand so that it can be placed wherever desired. The pump is substantially constructed, the workmanship being first class, and, as the parts are made to jigs and templates, they can all be easily renewed, being perfectly interchangeable. The ratchet wheel *D* and pawls *F* and *N* are made of tool steel, tempered and hardened. All other metal parts about the pump are made of the very best hard bronze composition.

Oil Electric Generating Plant.

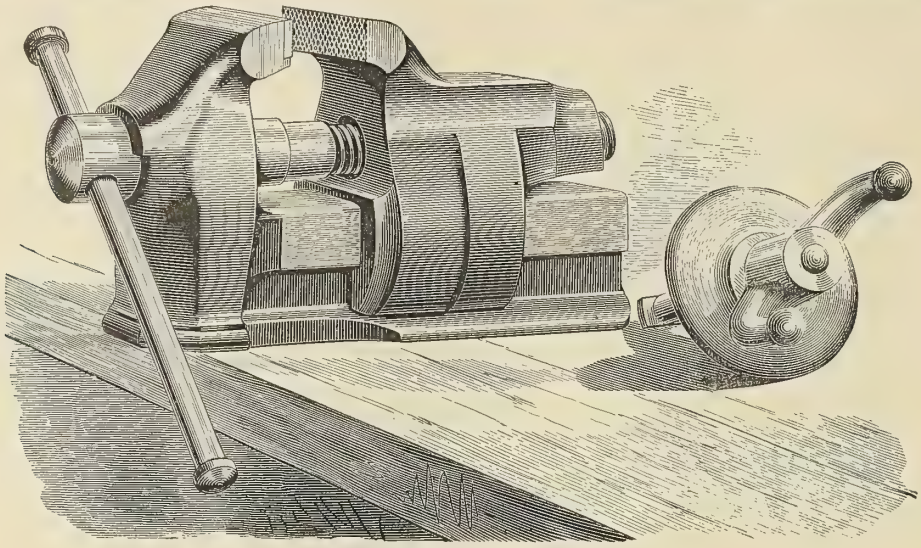
To produce electricity or any other power directly from fuel has been a problem for many engineers, and the resulting gasoline engine has given more or less satisfaction. For commercial use, however the gasoline engine is restricted because of heavy insurance rates. The other direct sources of heat are coal, natural gas, crude oil, or refined oil. The use of natural gas is, of course, limited to localities where it is produced; crude petroleum is difficult to volatilize and is not obtainable in every market. Up to the present much trouble has been encountered in using a refined oil such as kerosene, from the fact that an excess of oil or air in the mixture, and any variation in the temperature of the igniter, prevents continuous operation.



ELECTRIC GENERATOR DRIVEN BY AN OIL ENGINE.

The engine here illustrated, built by the General Power Company, 81 Fulton street, New York, N. Y., uses kerosene oil as fuel, and it is claimed that all the obstacles have been overcome, as the methods of supplying and igniting oil are radically different than those heretofore used. Instead of the oil being admitted to the engine as a vapor or forced in as a liquid, it is drawn in in the liquid state, and the exact mixture of oil and air is at all times obtained by means of a micrometric measuring device. Kerosene is superior to gasoline or city gas in chemical uniformity. In the design of this engine, simplicity, reliability, strength, and lightness have been aimed at. It is evident from the illustration that no excess material has been used such as is seen in a gas engine, the design being more like the high-speed steam engine. The cylinder, which is water-jacketed, rests on columns of open-hearth steel diagonally braced. The journals are unusually large and special provision has been made for copious lubrication. The cover of the cylinder and valve chest is made in single casting. The shaft is carefully balanced and the connecting rod is of the marine type. The cam shaft is actuated by a vertical lay shaft which is connected to the crank shaft below and the cam shaft above by helical gears, and the governor is placed in the top of the lay shaft by which it is operated, the object being to permit close connection between the governor and the air and oil inlet valves. Speed regulation is accomplished by varying the amount of oil and air taken into the cylinder and not by the usual hit-and-miss device. The engine may be started and stopped with as

much ease as a steam engine, and is controllable under all conditions. The regulation is so accurate that when the engine is direct connected to a generator a constant voltage may be maintained with all degrees of load; in fact, these engines have run very successfully alternators connected in parallel.



HEAVY CHILLED BEAM VISE.

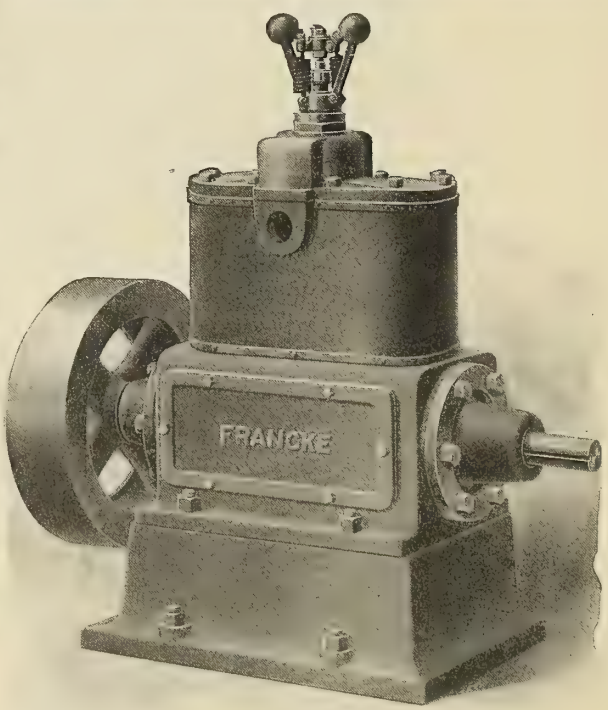
Heavy Chilled Beam Vise.

A vise which was designed and built specially to meet the requirements of machine-shop work has proved so satisfactory to the manufacturer in his own shop that it has been placed on the market. In its design and construction the following points were kept in mind: First of all, a vise must be sufficiently strong to allow of chipping or filing the work without the possibility of the vise breaking, and the inertia of the anvil should be sufficient to absorb the effects of the blows; the jaws should move parallel and freely, and when the screw is tightened the full power of the screw should be exerted on the work and not in overcoming the friction between the head and the movable jaw; all parts should be durable; and, lastly, the tool must be furnished at a reasonable price. These points are claimed to be embodied in the vise here illustrated. The jaws are steel faced by welding, and plenty of metal is used to secure strength in the proper place. The motion is direct and all sliding surfaces are chilled, thus parallel action is given, durability and increased strength secured. By making the vises with special machinery they are sold at a low price. They are made in five sizes, weighing from 15 to 125 pounds, with jaws of from 2 by 5-8 inches to 6 by 1 1-4 inches in dimension. For further particulars address the manufacturer, H. B. Smith Machine Company, Smithville, N. J.

Four-Ported Engine.

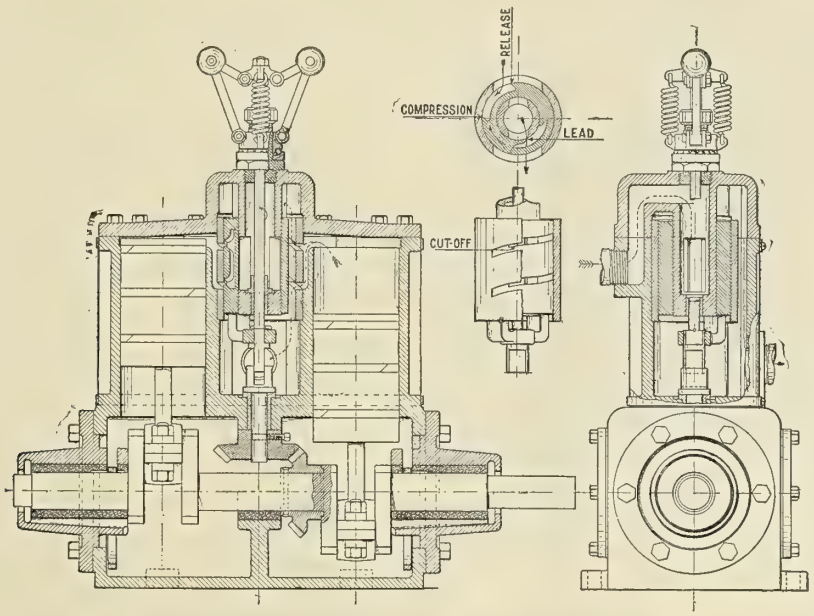
The Francke engine, herewith illustrated, is termed a four-ported engine, as it accomplishes with one valve steam distribution attained in a four-ported engine. The valve mechanism and engine are so simple in construction that the first cost is not higher than that of other first-class engines, while the highest grade of workmanship and material is maintained. The valve is simply a cylinder with a steam port on one side and an exhaust port on the other, turned on the outside to fit a bushing containing the ports leading to the cylinders, and on the inside to fit a tube containing the live steam ports. It is rotated by a pair of bevel gears from the shaft, running constantly in oil, and in all a very simple mechanism.

The valve is connected to the governor by means of a substantial spindle which drives the governor and makes the valve and governor practically one integral part, the valve rising and falling with the motion of the governor in a ratio determined by the relative



DOUBLE SINGLE-ACTING ENGINE WITH REVOLVING VALVE.

length of the governor arms. The steam port in the valve, one edge of which determines the lead given to the engine, is cut longitudinally with the valve on its inside diameter, so that the up-and-down motion imparted to the valve by the governor does not change the position of the valve with relation to the position of the piston, and the lead is maintained constant and may be changed at will without interfering with cut-off, release, or compression. On the outside diameter of the valve, the form of the port changes so that two ports are formed which have a slant corresponding to the slant of the ports cut in the bushing in which the valve rotates; consequently when the valve is raised by the action of the governor the cut-off takes place earlier, and when the valve is lowered it takes place later, without in any way changing the lead, release, or compression.



SECTION OF FOUR-PORTED ENGINE WITH REVOLVING VALVE.

The exhaust port in the valve is simply a slot cut longitudinally with the valve, one edge of which opens the port and the other edge closes it. As the valve edge which opens the port does not close it, which is the case in every reciprocal valve motion, it follows that the position of the opening edge may be set in exactly that position determined by the indicator to be the best for the speed at which the engine is run, without interfering with the lead, cut-off, or compression. Likewise the closing edge may be set to obtain the best compression for the economical use of steam, smooth running, and freedom from shock, which is so essential to high speeds, without interfering with the lead, cut-off, or release. Thus this single valve gives a four-valve distribution of the steam, which allows even the release and compression to be set independently of each other, which is impossible in even the best of the four-valve types. By referring to the cut it will be seen that the reciprocating parts of the conventional type of engine have been almost entirely eliminated, leaving only the two connecting rods and pistons, connected directly to the valve spindle itself, both the weight and the valve being driven by bevel gears from the main shaft, operating continually in oil. The governor's only function is to raise or lower the light-balanced valve according to the load of the engine. A close regulation is thus obtained with very light weights and springs and the simplest form of construction.

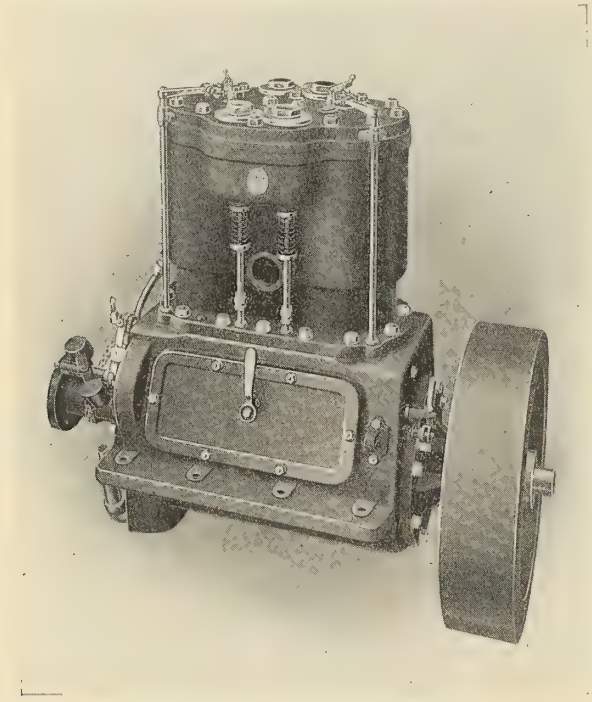
For lubrication, ring-oiling bearings are used for the shaft and an oil bath for the running gear in the crank case. A sight-feed lubricator takes care of the pistons and valves. The engine can be run at exceptionally high speed without undue wear or attention. Every engine is fully tested under steam before leaving the works, and is shipped ready to start. It is made in sizes from 5 to 25 horse power. This Francke engine is built in New Brunswick and sales are controlled by Thomas C. Maxfield, 120 Liberty street, New York, N. Y.

16-Horse-Power Four-Cycle Marine Engine.

The 16-horse-power four-cycle marine engine made by the Lozier Motor Company, Plattsburg, N. Y., embodies many late improved ideas in marine gas-engine construction, and presents many features of merit. One of these is, that all parts needing attention are accessible and at the same time the engine has a symmetrical appearance. These engines are of the medium-speed type. The inlet and exhaust valves are of the same size, thus interchangeable, and are mechanically lifted, which gives positive motion. The engine is so designed that any valve may be removed by simply unscrewing the cover which is set in the head over same, without removing any piping. The ignition mechanism is very simple and of the "hammer-break" type. The firing plugs occupy a central position in the cylinder head and thus can be easily removed. The valve chambers and other parts requiring to be cooled are water-jacketed, the water being circulated by a rotary pump which has proven to be very satisfactory and is driven by a chain. Cooling water is conducted from one cylinder jacket to another

through a covered passage, thus doing away with the unsightly pipes; in fact, the engine requires but little piping. The oiling is accomplished by sight-feed oil cups.

The motor is regulated by a governor which varies the lift of the admission valve, thus throttling the gases close to the cylinder. It will hold in constant speed with full or light load, and in heavy seas will prevent any tendency toward racing. When full power is wanted the governor may be easily disengaged and the engine will make fully 20 horse power. In addition to the automatic governor there are several other means of controlling and regulating the engine. The exhaust valves may be lifted by means of a lever to relieve the compression and permit the engine to run slowly or the flywheel to be turned by hand. The position of the igniter cam shaft is adjusted by a small hand lever placed at the front of the engine. The motor may be further controlled by means of a butterfly throttle placed between the vaporizer and the inlet valves.



16-HORSE-POWER MARINE GASOLINE ENGINE.

The material, construction, and workmanship are all high grade. The main bearings are bronze-bushed, and the cams, governing parts, and valve mechanism are case-hardened. The cylinder head for both engines is in one piece. The inlet and exhaust valves are on opposite sides of the engine, with a slight but equal overhang, giving a well-balanced appearance. The motor and reversing clutch are placed on one bed-plate, and next to this is the thrust-bearing bed, adjusted for go-ahead and astern motion.

Cruiser Contracts.—The Secretary of the Navy awarded contracts for building the armored cruisers *Washington* and *Tennessee* to the New York Shipbuilding Company, Camden, N. J., to Wm. Cramp and Sons' Ship and Engine Building Company, Philadelphia, the price being $4,035,000 each.

QUERIES AND ANSWERS.

Q. 122.—Please answer the following through the Marine Engineering:

1. What precautions are now taken to guard against galvanic action or corrosion on the interior of marine boilers?

2. What are the precautions taken to prevent scale deposits on the interior of boilers using salt water?

3. In using lake or river water in boilers what deposits are mostly to be feared and what precautions are taken to guard against them?

4. If you use well water in a boiler that is hard by reason of carbonate of lime or magnesia, how can it be made soft?

A. F.

A.—Corrosion is a somewhat broader term than galvanic action. Corrosion may be due to the chemical change known as oxidation or it may be due to the presence of an acid. Oxidation simply implies the chemical union between the metal and free oxygen. These two kinds of corrosion are, however, by no means independent, and are often combined in very complex ways. In order to understand galvanic or electro-chemical action a few explanations must be given.

Nearly all substances, and in particular metals and minerals, exist in a different electrical condition. Such a difference is often found not only between substances of different kinds, but also between similar substances at different temperatures or in different physical conditions, as, for example, between two pieces of iron or steel one of which has been heated or worked more than the other. Due to this difference of electrical condition there is a tendency to set up a flow of electricity from one to the other and at the same time to bring about certain chemical changes in connection with such an electrical current. The activity of such chemical reactions will depend very much upon the surroundings of the two bodies, and, in case they are immersed in a liquid chemically active toward one of them, there is great likelihood that the electro-chemical action will be brought about and will result in a continued destruction of one of the substances accompanied by a current of electricity around the circuit.

In a steam boiler the conditions for a more or less continued electro-chemical action may be fulfilled in a variety of ways. Parts of the structure of widely differing temperatures or of different physical or chemical compositions may provide the elements with a different electrical condition. Still more likely such a difference may be found between iron and its oxides, or between a particle of carbon in the steel and the surrounding metal, or between a bit of slag or other impurity and the surrounding metal. Copper also, either in the form of oxide or metal, would furnish a substance differing strongly from the iron. The exciting liquid is the water in the boiler, which, when salt and at high temperature, is capable of exercising a sufficiently marked chemical action to develop the electro-chemical reaction referred to above.

For the prevention of these chemical actions the causes must be either removed or neutralized. For reducing or preventing oxidation the presence of air in the feed water must be avoided by preventing, as far as possible, the entrance of water from overboard into the feed, or the possibility of the feed pump sucking air from the hot well. The piston rod of the low-pressure

cylinder should also be kept well packed, so as to prevent the entrance of air during the exhaust stroke. To prevent acid corrosion the formation of acids must be prevented, as far as possible, and such as may form must be neutralized within the boiler. The chief substances from which acids might arise are the salts contained in sea water or various organic substances. We have thus an additional reason for keeping such substances out of the boiler as far as possible. In order to neutralize such acids as may be formed, bicarbonate of soda, or soda ash, as it is commonly called, may be used from time to time as necessary. To test the water the usual litmus test may be employed. Blue litmus paper turns red when dipped in water slightly acid, while if the water is alkaline it turns it blue. Instead of soda, dependence is often placed on the zinc slabs referred to below and employed to prevent electro-chemical action. These are gradually dissolved by hot salt water, forming zinc chloride, and this will tend to neutralize any acids and keep the water alkaline.

In order to prevent electro-chemical action, the causes noted above must be removed or neutralized as far as possible. To this end the following conditions should be fulfilled: (1) Causes liable to produce oxidation or the formation of other foreign compounds should be kept out of the boiler; (2) the water in the boiler should be made as nearly neutral or non-exciting relative to iron as possible. This will be attained by keeping it slightly alkaline rather than acid and by avoiding high densities. In addition to this means for reducing the causes a further important step should be taken, and this is (3) the provision of a substance which shall be electro-positive to iron; that is, so related to iron that the electro-chemical action, if any takes place, will result in the destruction of the protecting substance rather than of the iron. Such a substance is found in zinc, and its use is quite general and apparently beneficial. The action of zinc is thus seen to be twofold. By the formation of zinc chloride it will tend to keep the water alkaline and thus reduce the chemical activity, while at the same time if there is electro-chemical action it will attack the zinc rather than the iron.

In order to be thus effective, however, care must be taken to insure actual metallic contact between the zinc and the structure of the boiler. The zinc is usually in the form of rolled or cast slabs weighing eight to twelve pounds each. These may be placed in perforated sheet-metal baskets hung from the stays or attached to other portions of the boiler. Where the basket is attached to the boiler there should be bright metal contact and the attachment should be by screw joints, so that the separation of the two surfaces by the formation of scale or corrosion may be prevented. The zinc should also be connected to the basket by through bolts or other means which will insure continuous metallic contact.

In dealing with questions of this character the principal point is to understand as clearly as possible the causes of the corrosion or other difficulty, and then to take such steps as will most certainly insure the removal of such causes or their neutralization at the point of action.

(The other points will be answered later.—Ed.)

Q. 123.—Will you kindly answer the following questions through your valuable paper:

(1) I am running feed water into the dome of a 5 by 7 Roberts boiler, through an economizer of my own make, at 260 degrees Fahr. If I extend my feed pipe down and back and forth over the fire and feed into the bottom leg (taking it out of the dome), will it work, or will it be better to feed in the dome at water line as usual?

(2) What is the best way to construct an ash siphon for lifting or blowing ashes out of the pit and overboard? G. W.

A.—(1) If you extend your feed pipe down and run it back and forth over the fire you make the feed pipe practically a part of the regular heating surface of the boiler, and it is doubtful if there would be any advantage in this. You might feed into the lower drum if you wish, direct from your economizer, although with this boiler and others of the water-tube type the usual arrangement is to feed into the upper drum. The usual arrangement of feeding from the economizer into the upper drum will probably prove most satisfactory, and we could hardly advise increasing the heating surface of the boiler proper in the way you suggest.

(2) As you doubtless know, there are two or three types of ash ejectors on the market which operate in the way you have in mind. A study of these will illustrate the points involved in the apparatus. Perhaps some of our readers may have had experience in fitting up in small size a simple device of this character, and if so we shall be glad to hear from them for the general benefit of those interested.

Q. 128.—Will you kindly inform me of the best way to deaden the sound of the exhaust of a two-cycle marine gasoline engine of four-horse power or larger? Is it possible to let the exhaust out under water, as some makers claim, similar to a four-cycle engine? Can the overflow water be allowed to intermingle with the exhaust in the muffler, both going out of the one outlet? L. A. C.

A.—One of the best ways to stop the noise is, as you suggest, running the exhaust under water, passing it first, however, through a muffler of from four to eight times the volume of the cylinder. This method is described in that chapter of Mr. Roberts' articles which treats on installation. Allowing a portion of the outlet water to pass into the exhaust pipe will help to deaden the noise, though not stop it entirely.

Q. 133.—Will you kindly answer the following questions. What are the government's requirements for the point at which a fusible plug should fuse? I have heard different fusing points. Also the area of steam and exhaust ports in proportion to size of cylinders. These questions are asked by inspectors for license. J. T. S.

A.—Answering the first part of your question, we would refer you to Section 25, Rule II of the Rules of the United States Board of Supervising Inspections of Steam Vessels, which is as follows: "The bronze casing of all fusible plugs, unless otherwise provided, shall have an external diameter of not less than that of a three-quarters of an inch gas or steam pipe screw tap, except when such plugs shall be used in the tubes of upright boilers plugs may be used with an external diameter of not less than that of a three-eighths of an inch gas or steam pipe screw tap (said plugs to conform in construction with plugs now authorized to be used by this board); and no such plugs shall be used unless the core of the plug is filled with good Banca tin from end to end of the bronze casing; and it shall be the duty of the inspectors to see that these plugs are so filled (with Banca tin) at each annual inspection."

The fusing temperature of Banca tin is about 455 degrees Fahrenheit.

As to the second part of your question, would state that the area of ports is determined by the speed of piston, the speed allowed the steam through the ports, and the area of the piston. The ports should be of sufficient size so that the steam will not be throttled in passing into the cylinder, resulting in "wire-drawing"; neither should the ports be too large, lest the clearance spaces to the cylinders be correspondingly large. It has been found that high-pressure steam will not flow as rapidly as low-pressure steam without resulting in wire-drawing. As the piston speed varies at all portions of the stroke, the mean piston speed is taken and we may express the quantities in the form of an equation:

$$V A = va, \text{ or}$$

$$a = \frac{VA}{v}$$

Where V = the number of strokes per minute × length of stroke in feet.

A = area of cylinder in square inches.

a = area of port in square inches.

v = velocity of steam through the port in feet per minute. This item is assumed, and, as above stated, is different for steam at different pressures. In practice it has been found that where the steam ports are used for exhaust as well as for admission, the velocity of the steam through the high-pressure port should not be more than about 5,800 feet per minute; the velocity through the intermediate port, 7,200 feet per minute; the velocity through the low-pressure port, about 8,600 feet per minute. Where a compound engine is used the above figures may be taken, only omitting that for the intermediate-pressure cylinder. These figures are good for ordinary practice, but where light high-speed engines are designed a higher speed is allowed for the steam passing through the intermediate- and low-pressure ports, say 8,600 and 11,500 respectively.

Q. 134.—Will you please answer the following question in your Queries and Answers column:

Supposing you were at sea on a steamer and the safety valve broke, also the steam gage. How could you tell the pressure on the boiler, and is there more than one way? G. C.

A.—The answer to a question of this kind depends entirely on the circumstances of the case, and on the outfit of spare parts, etc., in the store room. There should be a spare gage of some kind on board which could be attached in place of the broken one. If there were no such gage, a one-half inch pine connection might be connected to the boiler and the indicator attached with the heaviest spring in the outfit. Knowing the scale, this would serve to indicate the pressure. If there were several boilers in battery, and there was no occasion to work this one separately, there would be no particular need of a separate gage, and the pressure could be sufficiently shown by the gages on the other boilers.

It simply amounts to this: that to measure pressure you must have some suitable gage or instrument; and given such, it is simply a matter of attaching it suitably to the boiler in question.

Q. 135.—Will you kindly answer, in your valuable inquiry column, the following questions:

1. In case your main steam pipe broke off in flange, how would you bring ship in port?

2. How could I place a cylinder on frame, the lower end being rough? C. B.

A.—(1) To questions of this character there is no single or definite answer. Given an actual case with all its attendant circumstances and the resourceful engineer will usually find some way out of the difficulty; but with the simple statement that the steam pipe is broken in the flange we have before us only one item in a hundred which may affect the problem. Assuming that the flange alone is broken, and that it is separate from the pipe, it might be removed and a new one put on in its place, were such at hand. Again, it is perfectly possible that certain breaks might be repaired by a soft patch. Again, it might be perfectly possible to make a new flange from a spare piece of plate, or from a piece taken from the internal structure of the ship where it might be least missed.

A situation of this character can be readily analyzed down somewhat as follows: To get steam from the boiler to the steam chest a continuous and steam-tight pipe must be provided. You cannot ladle it in by the spoonful. If the pipe or connections are broken, repair may be effected in two ways, either by reinforcement or by substitution. Reinforcement means a patch or something of the kind. Substitution means a new piece. As to which of these is the more suitable, or as to how it is to be best carried out, can only be determined by a study of all the details of the case.

(2) It is not clear just what is intended by the question. If it means that the facing on the bottom of the cylinder where it is intended to rest on the frame is rough instead of smooth, then the problem is definite. As to how the cylinder could be placed on the frame, would depend on the means available for hoisting and handling such weights. As to how it should be secured there, which we assume to be the point of inquiry, the following suggestions may be made:

First make the facing on the bottom of the cylinder as smooth as practicable by chipping with a cold chisel. This will go far toward making a fit. If sufficient time is available for a good job here, nothing more will be required, and the fastening may be made by bolts in the usual manner. If there is not time to make a satisfactory fit, then a putty or cement of sal-ammoniac, sulphur, and iron borings, or glycerine, litharge, and iron borings, may be made and put on the top of the frame to fill in the open spaces. If these are large, steel wedges may be driven in to supplement the putty, and the bolt fastenings may be secured in the usual manner.

It is possible that the point of the question may have had reference to lining up a cylinder under the conditions specified. In such case the adjustment and lining up would proceed in the same manner and by the application of the same tests and by the use of the same appliances as in any case, and which for lack of space will not be described here without a further inquiry relating specifically to this operation.

Q. 136.—What is meant by the ductility of a boiler plate, and how is it determined? F. B.

A.—The ductility of a boiler plate is indicated by two results of the usual test which is made on the sample taken from the plate in question; chiefly by the *elongation*, and to some extent by the *reduction in area*. The percentage of elongation in 8 inches, the usual length of the sample between marks, is found by taking the total length between these marks just before rupture and subtracting from this the 8 inches or original length.

This gives the total elongation, and this divided by 8 and reduced to per cent. will give the percentage elongation. Thus suppose the original length of 8 inches is laid off, and just before rupture this has extended to 10.2 inches. The total elongation in 8 inches is then 2.2 inches. This divided by 8 gives 27.5 per cent. as the percentage elongation.

The percentage reduction of area is found by subtracting the final area of the section at the point of fracture from the original area at the same point, dividing the difference by the latter and reducing to per cent. Thus if the original area is .72 square inch and the final area is .38 square inch, the total reduction is .34 and the percentage reduction is .34 ÷ .72 = 47.2 per cent.

The United States requirements for boiler plate are as follows:

"The sample must show, when tested, an elongation of at least 25 per cent. in a length of 2 inches for thickness up to 1-4 inch, inclusive; in a length of 4 inches for over 1-4 to 7-16, inclusive; in a length of six inches for all plates over 7-16 inch thickness and under 1 3-4 inches thickness.

"All steel plate of 1-2 inch thickness and under shall show a contraction of area of not less than 50 per cent. Steel plate over 1-2 inch in thickness up to 3-4 inch in thickness shall show a reduction of not less than 45 per cent. All steel plate over 3-4 inch thickness shall show a reduction of not less than 32.5 per cent."

Q. 137.—On board the steamer *S*—— the boilers were filled with fresh water, and the first day out at sea, after steaming twenty-four hours, the water showed three-quarters at the salinometer. Does this necessarily mean that there is that much salt in it, or does it only indicate the density of water which might be affected from other impurities besides salt? B. F. S.

A.—Assuming the observation and the instrument both accurate, the result simply indicates that the density of the water is 3-4 on the salinometer scale. The presumption is that this is due to the salts in sea water, of which ordinary salt forms the larger part, or usually about 75 to 80 per cent. It would be rare that anything except sea water could affect the density of the water in the boilers to the extent mentioned, and in such a case we should be inclined to suspect the condenser of leaking. Such a condition might very readily cause a rise in density, as observed, and it is on the whole the most likely explanation of the situation noted.

Q. 138.—Would you do me the favor of explaining the accompanying sketch taken from the inset of the Scotch boiler in your issue of December, 1902.

Per cent. for rivets, Lloyds'.

Strength of shell, Lloyds'.

Or can you tell me where I can get a book of Lloyds' rules. Those figures don't agree with English B. T. Rules.

Plate section is correct.

U. S. Rules also.

Lloyds' (Haswell),

$$\frac{\text{Thickness} \times \text{Least per cent.} \times \text{Constant } 260}{\text{diameter shell in inches}} = P$$

$$\frac{23}{28}\left\{\frac{\text{Area rivets} \times \text{Rows} \times 1\ 3\text{-}4 \times 100}{\text{Pitch} \times \text{Thickness}}\right\} = \text{Per cent. Rivet Section}$$

T. R.

A.—Lloyds' rule for per cent. of rivet strength is:

$$E = \frac{n \times a}{p \times t} \times 85$$

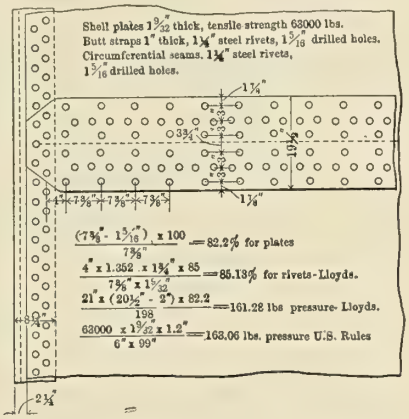
Where n = number of rivets.

a = sectional area of one rivet. When the rivet is in double shear use $1.75a$ instead of a.

p = pitch of rivet.

t = thickness of plate.

It is readily seen that $na \times 1.75$ gives the total sectional rivet area corresponding to one pitch, or p, while pt gives the corresponding sectional area of plate before


Shell plates 1 9/32" thick, tensile strength 63000 lbs.
Butt straps 1" thick, 1 1/4" steel rivets, 1 5/16" drilled holes.
Circumferential seams, 1 1/4" steel rivets,
1 5/16" drilled holes.
1 1/4"
3 1/2"
21"
1 1/8"
(7 5/8" - 1 5/16") x 100 / 7 5/8" = 82.2% for plates
4" x 1.352 x 1 1/4" x 85 / 7 5/8" x 1 5/32" = 85.13% for rivets-Lloyds.
21" x (20 1/2" - 2") x 82.2 / 198 = 161.28 lbs pressure-Lloyds.
63000 x 1 9/32 x 1.2" / 6" x 99" = 163.06 lbs. pressure U.S. Rules
2 1/4"

the holes are drilled. If the shearing strength of steel were the same as the tensile, then the numerator $na \times 1.75$ would represent the actual rivet strength in the same manner that pt represents the total original plate strength. Due to the fact, however, that the shearing strength is less per square inch than the tensile strength, the apparent rivet strength $na \times 1.75$ must be reduced to allow for this reduction in strength per square inch. Lloyds' rule assumes that the shearing strength is 85 per cent. of the tensile strength, and hence the numerator is multiplied by .85. The expression is then multiplied by 100 to give the answer in terms of per cent., and this brings the factor in the numerator to 85, as shown by the formula above. Comparison of this with the figures in the example quoted will show an entire agreement throughout.

Lloyds' rule for shell is as follows:

$$p = \frac{C\,(T-2) \times B}{D}$$

Where D = mean diameter in inches.

T = thickness in *sixteenths of an inch.*

B = lowest percentage for longitudinal joint, either for plates or for rivets, as determined.

C = factor depending on style of longitudinal joint and with values as follows:

21, when joint has double butt straps of equal width.

20.25 when joint has double butt straps of unequal width.

19.5 when joint is lap instead of butt.

In the present case the value of C is 21. The thickness of the shell is 1 9-32. This is equal to 41 thirty-seconds and hence to 20 1-2 sixteenths.

The value of T is therefore 20 1-2. The value of B, as previously found, is 82.2 for the plates, and the value of D is 198. The formula results, therefore, as shown by the figures in the example.

The formulæ which you quote are older forms of the rules. In the first the thickness is expressed in inches

and the constant is 2.60. This corresponds to a value $260 \div 12$ or 21.66 with the thickness expressed in sixteenths. In the second formula the quantities are the same, except that $23 \div 28$ is used instead of .85. The difference is small, $23 \div 28$ being equal to .821.

A copy of Lloyds' rules can be obtained from the New York office of the Society, Kemble building, Whitehall street, New York, N. Y.

Q. 139.—Can you inform me where I can purchase a good practical book on how to determine the proper size of paddle wheels of stern and side-wheel steamers for Western rivers? Also, if there is no such book published, would you please give through the columns of MARINE ENGINEERING the formula for determining the size of such wheels? J. C.

A.—We regret to state that there is no book published exclusively dealing with this subject. *Practical Marine Engineering* has a few pages on Western river practice, and Seaton and Roundthwaite's *Pocketbook for Marine Engineers* contains rules for determining sizes according to the English practice. From the latter we quote as follows from pages 185 to 187 for feathering paddle wheels:

Let V = speed of vessel in knots.

S = percentage of slip, usually 15 to 25 per cent.

R = number of revolutions per minute.

D = diameter of wheel to axis of float.

N = number of floats in one wheel.

K = a constant, which for vessels plying in smooth water = 1,200, for seagoing steamers = 1,400, and for other craft which are required to stop and start frequently = 1,600; then,

$$D = V \frac{100 + S}{3.14 \times R}$$

$$N = \frac{60}{\sqrt{R}}$$

$$\text{Area of one float in square feet} = \frac{I.H.P. \times 33{,}000 \times K}{N \times (D \times R)^3}$$

When a vessel works in smooth water the immersion of the top edge of the float should not exceed

$$\frac{\text{Breadth of float}}{8}$$

Q. 140.—Will you kindly answer the following questions for me:

(1) Fifteen pounds of steam at atmospheric pressure are led into an open barrel containing 15 pounds of water at the boiling point. What will be the resulting temperature? What weight of water will be contained in the barrel?

(2) Which engine will have the greater thermal efficiency, a non-condensing engine taking steam at 120 pounds gage pressure and exhausting it at 3.3 pounds above the atmosphere, or a condensing engine taking steam at 80 pounds gage pressure and carrying a vacuum of 25 inches? H. E.

A.—(1) In the first place it would be impossible to lead 15 pounds of steam at atmospheric pressure into an open barrel. The volume of one pound of steam at atmospheric pressure is 26.36 cubic feet, therefore 15 pounds would occupy 395.4 cubic feet. Assuming, however, that the barrel was large enough to contain this and the 15 pounds of water, which, as you state, is at the boiling point, there would be no change of temperature and no condensation except that caused by radiation. If the barrel were open, as you state, the steam would escape; but let us assume that the barrel is closed and the pressure not changed. The heat in one pound of water above 32 degrees at atmospheric pressure is 180.9 and the latent heat of steam at this pressure is 965.7. If any of the latent heat in the steam should be transferred to the water, the steam would be condensed and the water evaporated, resulting in the same condition as previously. What would happen, unless the barrel were steam-jacketed or the temperature maintained at 212 degrees to prevent radiation, would be that the steam would gradually condense until the steam and water would be at atmospheric temperature. However, total condensation would not occur until absolute zero was reached. If the air in the barrel had been entirely displaced when the steam was admitted and if the steam did not contain any air, this condensation would create a perfect vacuum at absolute zero temperature. The sum of the weights of water and steam in the barrel will at all times be 30 pounds.

(2) Referring to steam tables which will be found in most engineering hand books, the total heat of steam at 120 pounds gage pressure is 1188.6, and at 3.3 pounds is 1149.8. From the same tables we also find that at 80 pounds gage steam pressure the total heat is 1180 6, and at 25 inches vacuum is 1122.5. Therefore, the thermal efficiency, regardless of radiation or cylinder condensation, is as follows:

		Non-Condensing.	Condensing.
a	Admission pressure	120 lbs.	80 lbs.
b	Total heat of steam at corresponding pressure	1188.6	1180.6
c	Exhaust pressure	3.3 lbs.	25 in. vacuum
d	Total heat of steam at exhaust pressure	1149.8	1122.5
e	Loss=*a*—*d*	38.8	58.1
	Thermal efficiency=*e*÷*a*	3.27	4.92

Thus the thermal efficiency of the condensing engine is about 50 per cent. greater than that of the non-condensing engine. The reason of this is that there is not as great a change in the heat of steam at a difference of, say, 5 pounds at high pressures as there is at a corresponding difference at low pressures. Other than thermal considerations, however, bring the efficiencies of the non-condensing and condensing engines to within about 25 per cent. of each other, in favor of the former type.

Q. 141.—Would you please state briefly through the MARINE ENGINEERING what are the advantages and disadvantages of multiple-expansion marine engines and high steam pressures? F. P.

A.—A general discussion of this question would require far more space than can be here afforded, and we can only briefly outline the chief points involved. The question is really twofold: What are the advantages and disadvantages of high steam pressures, and what are the advantages and disadvantages of multiple-expansion engines?

Regarding the first question, the advantages can all be comprised under the one head of increased economy. If the high-pressure steam undergoes what may be termed full expansion, then the expansion ratio will increase with the increase of pressure, the range of temperature through which the engine works will be increased, and the coal economy will be improved. That is, a horse power will be obtained for less coal than with lower pressure steam. On the other hand, if it be desired to sacrifice coal economy, then the expansion ratio can be reduced, the mean effective pressure increased, and the size of the engine for a given power decreased. This will result in an increased weight economy. High-pressure steam can, therefore, be made to give either greater coal economy or greater weight economy, or, in fact, any combination desired between the two.

The disadvantages may be summed up under the gen-

eral head of greater care and greater wear and tear. High-pressure steam is of higher temperature, requires more carefully-packed joints, may give trouble with matters relating to lubrication, and, in general, requires more careful use than low-pressure steam. In none of these points, however, is the difficulty serious, and modern engineering methods are fully capable of taking care of steam up to, say, 300 pounds pressure, or covering the range of the prevailing practice of the day.

Regarding the second question, it results from the action between the cylinder walls and the expanding steam that economically only a very moderate degree of expansion can be carried out in any one cylinder, and that in consequence the total expansion must be broken up and carried on in a series of cylinders, in order to take full advantage of the possible economic gain. This loss of economy in a single cylinder is due to the excessive condensation of the steam by the cylinder walls, where any considerable degree of expansion is carried on in a single cylinder. To avoid this the operation must be split up and divided among a series of cylinders, and it thus results that the multiple-expansion engine is a necessity if full advantage is to be taken of high-pressure steam with regard to improved coal economy.

On the other hand, for marine purposes a multiple-crank engine is a practical necessity for mechanical reasons. It gives a more uniform turning moment on the shaft, provides for a more accurate balance of inertia forces, and avoids the difficulty in starting and handling which would be met with in a single-crank engine.

These various features of a multiple-expansion engine are so much a necessity that the other characteristics which might be urged as disadvantages have but little relative weight. It is true that a multiple-expansion engine has more parts, weighs more, costs more, and occupies more space than a simple engine of the same power; but for marine purposes in particular all these disadvantages are far more than outweighed by the advantages previously noted. So long as the reciprocating engine is the accepted marine type, it seems likely that it will be of the multiple-expansion form, using as high-pressure steam as the particular conditions justify.

United States Rules for Boilers.

Editor Marine Engineering:

In your Q. and A. columns in January issue I was interested in question 120, pertaining to boiler design. The question concerning thickness of shell gives factor of safety of 4 1-2 and strength of joint 80 per cent., also 1-16 inch to be added to shell for corrosion. The subsequent calculations are based on the United States Rules. While the question has been answered correctly, the premises in the shell computation are radically different from those in the United States Rules, in so far that the latter do not consider strength of joint, and their factor of safety is 6 plus 20 per cent. added to pressure for double riveting, practically making 5 the factor of safety in this case, and the shell thickness would then figure only 0.90 inch, omitting the 1-16 inch added for corrosion.

This addition for corrosion will, according to the United States Rules, incur a severe penalty, as, by a recent ruling of the United States Board of Supervising Inspectors, they have revived a dead letter law (Sec. 5, page 22), "Where flat surfaces exist . . . the spacing and distance apart of bracing and all other parts of the boiler must be so arranged *that all will be of not less strength than the shell.*"

Therefore, in Q. 120, if a boiler 12 feet diameter, tensile strength 60,000 pounds, and double riveted, be given a shell 1 1-16 inches thick, the calculation by United States Rules will give 177 pounds as the working pressure; and furnaces, stays, and all other parts will have to be designed for 177 pounds, even if it is desired to carry only 150 pounds working pressure. Of course the increase in size and weight will increase the cost.

Reviving the above-mentioned law has caused considerable comment and aggravation latterly, and constructors, engineers and boiler makers are anxiously looking forward to its repeal. H. L. E.

REMARKS.

The answer to the question referred to was framed to correspond to the terms of the question itself, and it was well known that the basis assumed in the question was quite different from that furnished by the United States Rules. As noted, the factor of safety in the latter for double riveting and drilled holes is 5 instead of 4 1-2, and it is rare to add such an allowance as 1-16 inch for corrosion.

The recent revival of the old rule referred to seems to be quite uncalled for, and, in the case of a boiler built under United States Rules, would severely penalize such a shell as that referred to in the question noted. We are indebted to our correspondent for noting this revival of an obsolete rule, and would join in the hope for its repeal by the Board of Supervising Inspectors.

Boilers of the British Navy.—The interesting results of the official steam trial of vessels of the British navy are given in *Engineering*, London. Six battleships, seven first-class cruisers and three twin-screw sloops, and two torpedo boats have undergone their steam trials during the year 1902. The battleships and cruisers are all fitted with Belleville water-tube boilers. The results of the 8-hour full-power trials on the battleships were that an indicated horse power was developed on one from 1.95 to 2.4 pounds of coal per hour; on the cruisers the average coal consumption on full-power trial was less, ranging from 1.82 to 2.12 pounds coal. Of the sloops the *Nerlin* is also fitted with Belleville boilers, and the pounds of coal burned per indicated horse power on a similar trial were 1.64 pounds; the *Odin*, the second of the three sloops, is fitted with the Babcock and Wilcox boilers, and the consumption was 1.57 pounds coal; the third sloop, *Fantome*, has Niclausse boilers, the results were 1.86 pounds of coal burned per indicated horse power. Thus it will be seen that the Babcock and Wilcox boilers give the best results of any of the types. The torpedo boats are supplied with Clydebank-Normand and Thornycroft boilers; the coal per I. H. P. was 2,455 and 2.62 pounds respectively. The Babcock and Wilcox boilers are to be installed on 11 vessels of the British navy, including 5 battleships, 4 first-class cruisers, and 2 second-class cruisers, with a total of 170,000 horse power.

The Burned Steamer Columbia.

One of the finest equipped passenger packet boats on the Monongahela—the *Columbia*—owned by the Pittsburg and Brownsville Packet Company, was totally destroyed by fire on January 16, 1903. The *Columbia* came in to the wharf of the company in the morning and was due to leave at four o'clock that afternoon. At two P.M. fire broke out and the flames quickly spread from end to end and destroyed everything above deck.

The cause of the fire is unknown, but it is presumed to be the result of boiling tar catching fire. The fire was a most spectacular one, and was witnessed by thousands from the bridge, shore, and hillsides. The fire also damaged the wharf boat to which the *Columbia* was moored. The vessel was built at Brownsville in 1901 and was valued at $50,000. The machinery and hull may be repaired, but the loss will amount to $25,000 and is not covered by insurance.

MONONGAHELA RIVER STEAMER COLUMBIA, BURNED AT PITTSBURG.

PERSONALS.

Mr. Chester D. Hubbard, who had partial charge of the electrical equipment at the Newport News Shipbuilding and Dry-Dock Company, will take complete charge of the electrical plant and the equipment of ships at the Bath Iron Works, Bath, Me.

The firm of Herreshoff and Wells, Naval Architects and Engineers, New York, N. Y., has been dissolved and Mr. Theodore D. Wells, of that firm, has gone into partnership with Mr. H. C. Wintringham, Naval Architect, 32 Broadway, New York, N. Y.

William Stevenson has been made chief draftsman of the hull department of the Townsend-Downey Shipbuilding Company, Mariner Harbor, New York, N. Y.

At the annual meeting of the Marine Engineers' Beneficial Association, held at Washington in January, all the officers were re-elected for the ensuing year. They are as follows: President, Geo. Uhler; vice-president, Frank A. Jones; second vice-president, Evans I. Jenkins; secretary, Geo. A. Grubb; treasurer, Albert L. Jones; advisory board, Joseph Brooks, John McG. Sterritt, William Scheffer.

Austin E. Overman, who formerly held a position as draftsman with the William R. Trigg Company, Richmond, Va., has taken a position in the drafting room of the New York Shipbuilding Company, Camden, N. J.

James A. Bimson has been appointed chief draftsman of the engineering department of the Bath Iron Works.

Guy A. Jackson, formerly in charge of the construction of the Standard Oil Company barge *No. 93*, at A. Sewall Company, Bath, Me., has accepted a position with the Eastern Shipbuilding Company, Groton, Conn., as foreman on one of the large passenger boats under construction there.

Lieutenant Commander F. C. Bowers, U. S. N., succeeds, at the Bath Iron Works, Rear Admiral Harrie Webster, who was retired this month.

Wesley B. Smith, former quartermaster at the Wm. Cramp and Sons' Ship and Engine Building Company, Philadelphia, Pa., has accepted a position as master steam fitter at the works of the Fore River Ship and Engine Company, Quincy, Mass.

TECHNICAL PUBLICATIONS.

Marine Almanac, 1903. Edited by Mittheilungen aus dem Gebiete des Seewesens. Size 3 1-2 by 5 1-4 inches. Pp. 588, with marginal index, blank sheets for notes, and many illustrations. Gerold and Co., Vienna, Austria. Price 4 1-4 marks.

The constant growth of naval armaments calls for an increasing dependence on statistical information, and among the various hand books prepared with this end in view, none perhaps stands higher than the present publication, having reference to accuracy, condensation, and convenience of size for ready reference. It is prepared yearly under the immediate auspices of the Austrian Department of Marine, and, while primarily intended for German-speaking peoples, it has justly become a standard book of reference for all topics relating to the subjects of which it treats.

Aside from the calendar and other miscellaneous information, the subject matter proper is given under five divisions.

Part I contains a collection of tables of weights and measures as used in various countries, with tables for the reduction of English measures to metric and *vice versa*. For the English reader these tables will be of value as furnishing a ready means of converting the various data given elsewhere in the book into measures with which he is familiar.

Part II consists of a compendium of laws and regulations relating to the Austrian naval personnel and to the general administration of Austrian naval affairs. This section will be of interest only to those directly concerned, or to the general student of naval administration in its minor details.

Part III gives in tabular form the usual statistical matter relating to the naval ordnance of the various maritime nations.

Part IV gives in tabular form the usual matter relating to the characteristics and leading particulars of the ships belonging to the various naval powers. The tabular matter covers 162 pages and is followed by 139 pages of line drawings giving in semi-diagrammatic form representations of the character and distribution of the armor and battery. This in turn is followed by a squadron list of the various naval powers and by a general index of all ships included in the section. This part of the book forms the section of greatest value and general interest, and seems to have been prepared with the usual accuracy and care.

Part V contains the Austrian navy list and has in consequence no especial interest for the general reader.

This little hand book of naval information has gained an assured place in the literature of the subject and it may be cordially commended to all interested as an accurate and authoritative collection of data and general information relating to the ships and armaments of all naval powers.

Steam Power Plants, Their Design and Construction. By Henry C. Meyer, Jr., M.E. Size 6 by 9 1-2 inches. Pp. 160. With 65 illustrations. Price $2. The McGraw Publishing Company, 114 Liberty street, New York, N. Y.

This book is written especially for engineers and others in charge of power plants, who may be called upon to design or purchase a plant or parts of it when their knowledge of machinery is more or less limited and when they are unable to obtain the advice of a consulting engineer. The book is a revised reprint of a series of articles which appeared in the *Engineering Record*, and the illustrations, most of which are insets, are taken from cuts which were published in that periodical. They represent a great variety of designs of power plants and many of the largest ones in the country. The pages tell of the designing of steam-power plants, of propellers and designs of boilers and engines, and sets of specifications for same. The subjects of steam and water piping auxiliaries, including feed-water heaters and economics, are dealt with, and the concluding chapters take up mechanical drafting, construction and design of chimneys, and problems of cold-water supply and purification. It is not claimed by the author that the services of a competent engineer should be dispensed with in designing a new plant by the use of this book, but so many plants are designed without such professional talent that the book is especially recommended to those of this class. To the expert the book claims only to present in accessible and convenient form data that are useful in his profession. The text is explanatory of the various problems met with in power-plant construction, steam-engine designing, and gives rules and formulæ for determining sizes and capacity of units. The illustrations form a valuable record of what has been done in this line, and, altogether, the book is a valuable one for all interested in power plants.

The Steam Turbine. By Robert M. Nielson. Size 5 by 8 1-2 inches. Pp. 163, with 145 figures in the text and 9 full-page plates. Longmans, Green and Co., London and New York. Price $2.50.

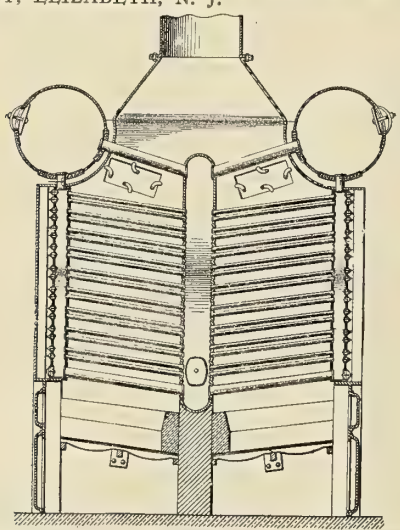
The author believes that the steam turbine is likely to be used to an increasing extent in the future, and that it has already assumed an importance which may well justify the preparation of a work dealing with the subject. He has therefore endeavored to combine in one connected treatment some discussion of the steam turbine historically, theoretically, and practically, and in the latter connection it has been his purpose to describe not only the principal parts of the leading types of steam turbine, but also the small details which in such cases have so large an influence in determining the success or failure of the turbine as a whole. As an aid to the discussion of the theory of the steam turbine from the thermodynamic point of view, a chapter has been added on temperature-entropy diagrams and their application to such motors. The results of a large series of tests are also given, care having been taken to make the values given as accurate as possible.

As the first extended book on the steam turbine this volume doubtless heads a long and growing list of future works. The fact that the turbine has come forward to a place meriting the recognition of a special book is an index of the growth during the past few years in its status as a prime mover. The author has a most timely subject for a new and interesting discussion, and he has well improved his opportunities. The style is good, and the book, without pretending to give any extended discussion of the theory of the turbine, does nevertheless give such a combination of the theoretical and practical as will be most useful to the general engi-

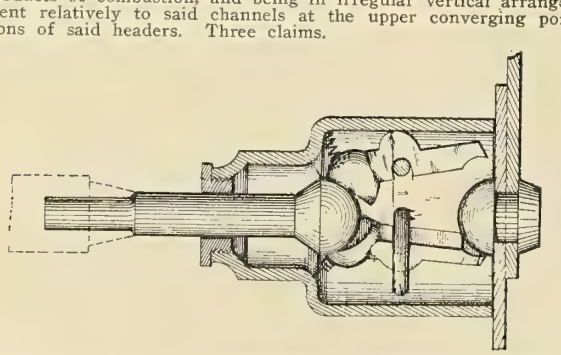
neering reader who is interested in this subject. Regarding application to marine propulsion, the marine engineer will perhaps wish for relatively more space. Eleven pages is an unduly small part of the book in which to discuss the special features of this field of application. While the leading types of turbines are supposed to be described, the Curtis in the United States, as represented on the yacht *Revolution*, is not noted and possibly was not known. Naturally the Parsons and De Laval came in for chief mention, or more particularly the former, as representing the type best known to the author and regarding which data were more readily obtainable. The book may be commended to all interested in the steam turbine as a prime mover, and its careful perusal will give a very excellent idea of the present condition of the turbine as a machine, and a good understanding of the principles on which it is operated and through which improvements must be sought.

SELECTED MARINE PATENTS.

714,530. WATER-TUBE STEAM BOILER. PATRICK J. SWEENEY, ELIZABETH, N. J.



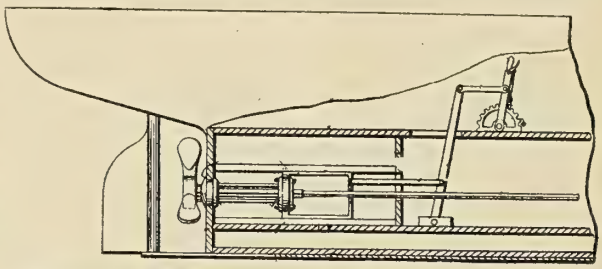
Claim.—1. In a water-tube steam boiler, the combination, with a rectangular casing, having two opposite sides, converging at the top and provided with doors to form the sides, of front and rear headers, an intermediate header having its interior and exterior surfaces corrugated, inclined water tubes connecting said headers, said headers converging at their upper portions in conformity with said casing, fire chambers below said water tubes, said water tubes being in regular vertical arrangement at the lower portions of said headers, to form channels for the products of combustion, and being in irregular vertical arrangement relatively to said channels at the upper converging portions of said headers. Three claims.



717,624. RIVET-CALKING TOOL. PATRICK J. SWEENEY, ELIZABETH, N. J.

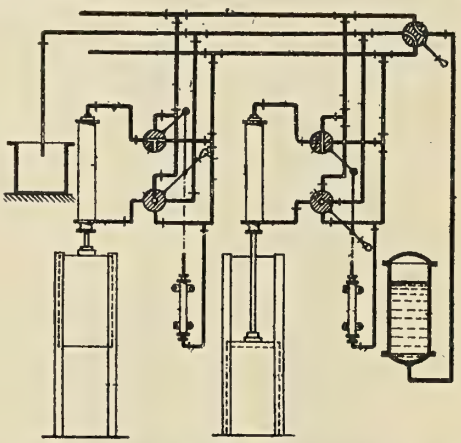
Claim.—1. A rivet-calking tool, comprising an exterior shell or casing, open at its lower end for receiving the rivet to be calked, a plurality of movable calking hammers in said casing arranged to receive the rivet head between them, and means for simultaneously operating said calking hammers into and out of contact with said rivet head, substantially as set forth. Four claims.

718,881. PROPELLER WHEEL. ORA E. SARR, CEYLON, OHIO, ASSIGNOR OF ONE-HALF TO DELLA SARR, CEYLON, OHIO.



Claim.—1. In a screw propeller, the propeller shaft, spaced propeller blades movably supported upon said shaft, spaced tracks parallel to said shaft, a frame slidably disposed between said tracks and enclosing said shaft, a hub slidable upon said shaft and revolubly disposed within said frame, rods connecting said hub and said propeller blades, and means for adjusting said frame and hub longitudinally of said shaft, whereby the pitch of said blades will be simultaneously and correspondingly altered. Eleven claims.

718,973. MEANS FOR OPERATING BULKHEAD DOORS. ALARD DU BOIS-REYMOND, BERLIN, GERMANY, ASSIGNOR TO NORDDEUTSCHER LLOYD, BREMEN, GERMANY.



Claim.—1. In an arrangement for simultaneously operating several bulkhead doors the combination with a common controlling valve, of a hydraulic cylinder operatively fitted to each door, means for connecting one end of said cylinder with either of a pair of mains leading to the controlling valve and the opposite end of said cylinder with either of the said pair of mains or with an exhaust pipe.

2. In an arrangement for simultaneously operating several bulkhead doors the combination with a common controlling valve, of a hydraulic cylinder arranged to operate each door, pipes leading from either end of said cylinder, and a pair of valves connected to said pipes and adapted to switch one of said pipes onto either of a pair of mains leading to the controlling valve and the other of said pipes onto either of the said mains or onto an exhaust pipe. Nine claims.

719,002. COMPOUND FOR REMOVING SCALE FROM STEAM BOILERS. JOHN N. HENDERSON, NEW ORLEANS, LA., ASSIGNOR TO EDWARD CUNNINGHAM, NEW ORLEANS, LA.

Claim.—A compound for removing scale from steam boilers composed of the following ingredients; parched ground coffee, extract of logwood, blood meal, salt, all thoroughly mixed with water. One claim.

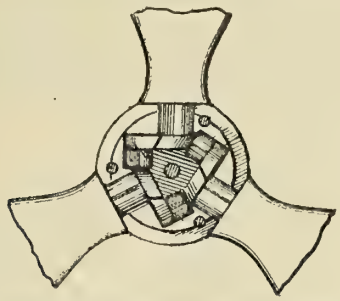
719,172. PROPELLER. ARTHUR DE BAUSSET, NEW YORK, N. Y., ASSIGNOR TO HIMSELF AND CHAR A. McCREADY, NEW YORK, N. Y.

Claim.—1. The combination with a shaft and propeller mounted thereon, of an open-ended cylinder receiving the propeller and in which the same is adapted to be rotated, a cylinder surrounding the aforesaid cylinder with an appreciable annular space between the cylinders, the outer cylinder being open at one end and closed at the other end, bearings for the said shaft, and devices maintaining the cylinders in a fixed relation to one another. Two claims.

719,225. REVERSIBLE PROPELLER. JOHAN V. JOHANSSON, SKRYSTA, HAMMAR, SWEDEN.

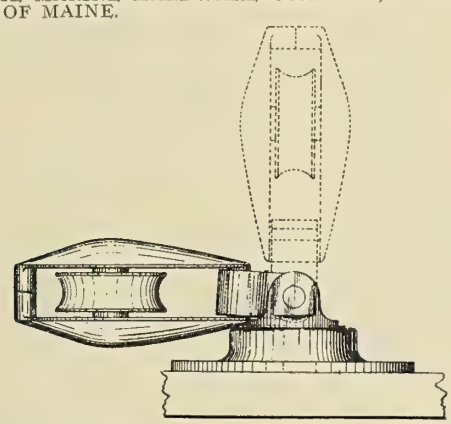
Claim.—1. In a propeller the combination of a shaft, a slide fastened thereto, said slide having a plane surface with a flange projected from one side thereof, the slide also having a transverse slot therein, a propeller blade, a quadrant on said blade and lying on the said plane surface of the slide, a pin attached to the quadrant and playing in the slot of the slide, the side edges

of the quadrant being arranged to bear against the flange of the



slide as the blade is thrown fom one position to another, and a hub in which the blade is mounted to turn. Two claims.

719,124. SHEET LEADER BLOCK. WILLIAM H. MARSHALL, WAKEFIELD, MASS., ASSIGNOR OF ONE-HALF TO THE MARINE HARDWARE COMPANY, A CORPORATION OF MAINE.



Claim.—1. In a sheet leader block the base-plate provided with the substantially vertical annular flange *e*; a plug rotatably supported by the base-plate within said flange; a sheave frame and pulley; and a pivotal connection between the plug and the sheave frame. Two claims.

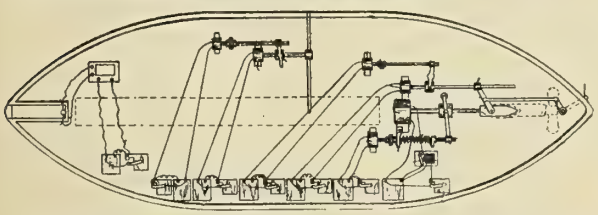
719,235. VENTILATION OF SUBMARINE BOATS. SIMON LAKE, BRIDGEPORT, CONN.

Claim.—1. In a submarine boat, the combination with a ventilating tube or passage communicating with the interior of the latter and opening outwardly above the same, of an outwardly-opening, spring-pressed check valve in said tube or passage located below its outer end, a chamber permanently open in its upper portion being formed above the seat of said valve and adapted to retain water entering the same, whereby the weight of the water serves to press said valve to its seat and to automatically hold it closed. Four claims.

719,316. INFLATING DEVICE FOR LIFE BELTS. JOHN EWING, JR., RICHMOND, CANADA.

Claim.—1. In an inflating device for collapsible life belts, the combination with the gas-generating chamber having a suitable opening thereinto, of a stopple having an eyelet or staple attached to its under side and designed to close the said opening, a bag or tube of porous material closed at each end, and designed to contain a gas-producing constituent, a string attached to the bag at one end and to the said staple at the other. Five claims.

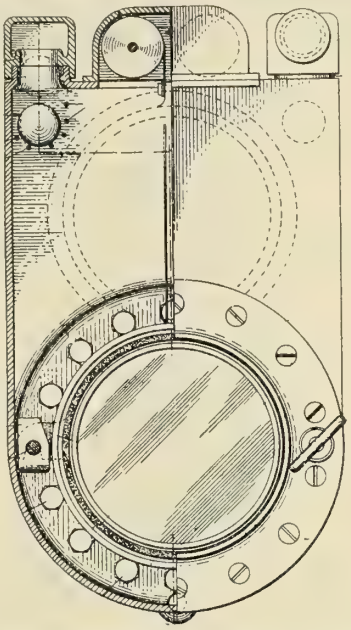
719,405. APPLIANCE FOR ELECTRICALLY OPERATING VESSELS AND TORPEDOES FROM A DISTANCE, LIDA WILSON, BROOKLYN, N. Y.



Claim.—1 The combination, with a movable vessel, of a series of electric motors for operating the propeller, the steering gear, the ballast shifter, and other required agencies, an electric battery or batteries upon the vessel for operating such motors, a circuit and circuit-closing mechanism connecting each of such motors separately with the battery, a series of wireless receivers of electricity each having means actuated thereby and connected with one of the circuit-closing mechanisms, and a corresponding series of wireless transmitters of electricity, such receivers and transmitters being mutually sympathetically synchronized in unison in pairs, whereby any of the said motors may be operated at pleasure from a convenient distance. Four claims.

719,416. VENTILATING PORT FOR VESSELS. JOSHUA W. ATLEE, RIVERTON, N. J.

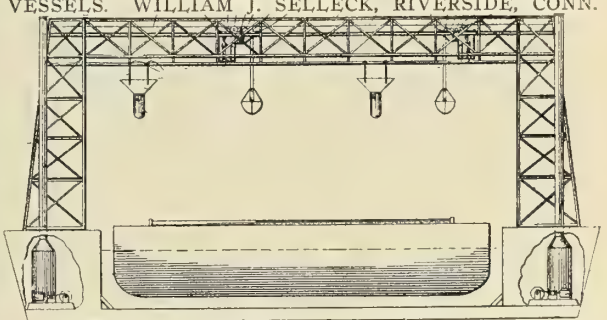
Claim.—1. In a ventilating port for vessels, a casing comprising inner and outer plates having openings to admit light and



air, a glass or cover adapted to close one of said openings and be retained in position within said casing at its upper portion when not in use, openings in the upper portion of said casing for the admission of air and an opening at the lowest portion of said casing communicating with the exterior of the vessel and adapted to carry off entrapped water.

2. In a ventilating port for vessels, a casing comprising inner and outer plates, having openings to admit light and air and being extended upwardly to form a receptacle for the glass when not in use, a glass or cover adapted to close one of said openings, secondary openings in the upper corners of said casing and float valves retained in said corners and adapted to close said openings to prevent the ingress of water. Eight claims.

719,517. APPARATUS FOR LOADING OR UNLOADING VESSELS. WILLIAM J. SELLECK, RIVERSIDE, CONN.

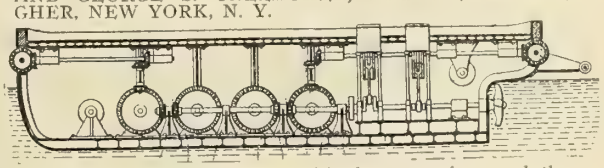


Claim.—1. In an apparatus, the combination with a floating vessel, of a bridge extending longitudinally above the vessel, a traveling crane guided on said bridge, and buoyant supports beneath the ends of said bridge, located in alignment with the bow and stern of the vessel. Thirteen claims.

719,654. MARINE LIFE-PRESERVER. JOHN EWING, JR., RICHMOND, CANADA.

Claim.—1. A marine life-preserver comprising a flexible casing adapted to be inflated, a partition dividing off a part of said casing to form a gas-generating chamber, said partition having a gas passage therethrough, and a cap closing an opening to said gas-generating chamber, said cap being adapted to admit water and to prevent the escape of gas. Five claims.

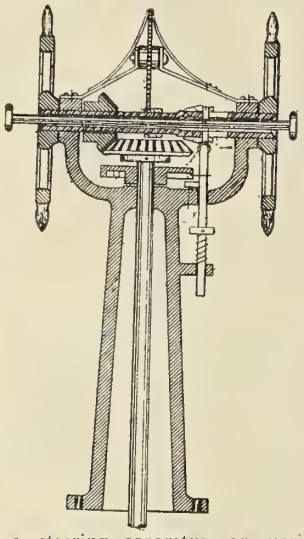
719,907. CANAL-BOAT PROPULSION. JOSEPH C. TONE, IRONDEQUOIT, N. Y., ASSIGNOR TO HIMSELF AND GEORGE S. GALLAGHER, AND HENRY GALLAGHER, NEW YORK, N. Y.



Claim.—1. The combination with the power boat and the engine thereof, of a series of large sheaves or rollers in line longitudinally of the boat with their axial centers in substantially the same plane, and devices substantially as shown and described for rotating the said series of sheaves simultaneously

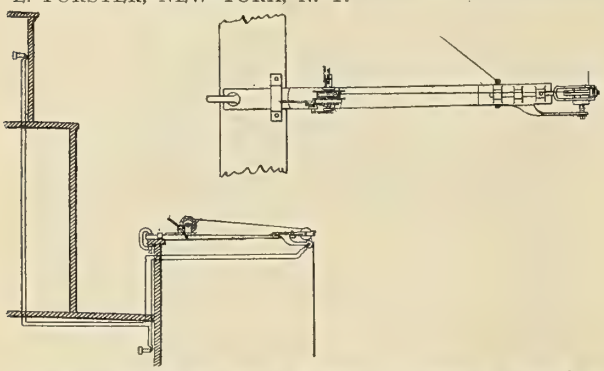
and in two sets in opposite directions by the engine, and a rope or chain passing over and alternately under the respective sheaves. Four claims.

719,869. STEERING APPARATUS. DANIEL W. RANTINE, BROOKLYN, N. Y.



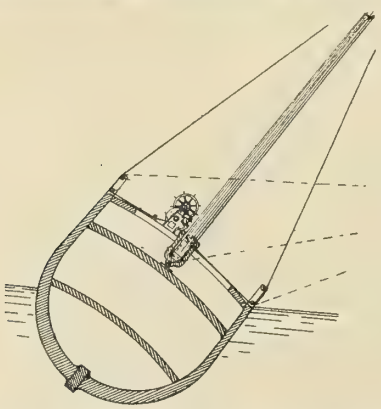
Claim.—1. In a steering apparatus, an upright support, a tiller-shaft passing vertically therethrough and provided at its upper end with a beveled gear wheel, a tubular shaft mounted horizontally thereover and provided with a corresponding beveled gear wheel of less diameter than that on the tiller-shaft, a hand wheel connected with said tubular shaft, a rod passing longitudinally through said tubular shaft and movable therein, a locking disk connected with the tiller-shaft, and devices operated by said rod and operating in connection with said disk for locking the tiller-shaft against rotation. Five claims.

720,015. SHOAL-WATER INDICATOR. FERDINAND E. FORSTER, NEW YORK, N. Y.



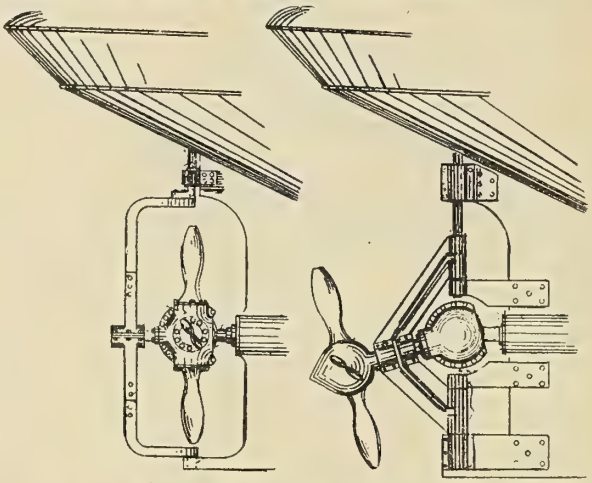
Claim.—1. In a shoal-water indicator, a sounder, adapted to be submerged and travel with the vessel, a sustaining wire attached to said sounder, and a resonator actuated by the vibrations of said wire and adapted to indicate audibly the contact of said sounder with the bottom. Ten claims.

720,074. AUTOMATICALLY LOWERING MAST. CARL SCHWANEBECK, GLUECKSTADT, GERMANY.



Claim.—1. The combination of the mast, the socket in which it is stepped, hinged to swing athwart the ship, the shrouds, the rollers in the bulwarks around which the shrouds are led, and the windlass to which the ends of said shrouds are connected in such a manner that while those on the windward side are paid off, those on the lee side are proportionately taken in. Five claims.

719,958. SCREW PROPELLER FOR THE PROPULSION AND STEERING OF SHIPS. JOHN SINCLAIR, BALMAIN, NEW SOUTH WALES, AUSTRALIA.



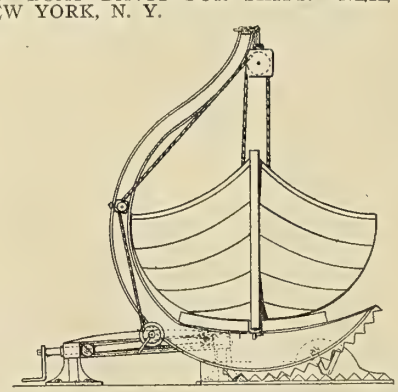
Claim.—1. A main or propeller shaft, a spherical hub on said main or propeller shaft, a divided sphere having a segmental opening embracing said hub, the said hub having two diametrically opposite driving pins and said divided sphere being slotted to receive said pins, both ends of slots being closed, and propeller blades mounted and balanced directly on said dividing sphere. Six claims.

720,076. ATTACHMENT FOR RANGE-FINDERS. ROBERT J. SHEEHY, NEW YORK, N. Y.

720,101. TORPEDO FOR SUBMARINE MINES. FRITZ BAUDUIN, LA HAYE, NETHERLANDS. Six claims.

720,508. ELECTRICAL SHIP'S-LOG APPARATUS. THOMAS F. WALKER AND THOMAS S. WALKER, BIRMINGHAM, ENGLAND. Two claims.

720,274. BOAT DAVIT FOR SHIPS. NEIL MURCHISON, NEW YORK, N. Y.



Claim.—1. A device for the purpose specified, comprising the toothed tracks, the connected davits having convex, toothed bases mounted on the said tracks, the top beam of the davits, the chocks mounted on the convex bases of the davits, tackles for suspending the boat from the top beam, and means for rocking the davits and the chocks on the tracks. Four claims.

Six Months' Shipbuilding Returns.—The Bureau of Navigation reports 627 sail and steam vessels, of 171,599 gross tons, built in the United States and officially numbered during the six months ended December 31, 1902.

Lehigh University.—A course in marine engineering is given by the Department of Mechanical Engineering at Lehigh University, South Bethlehem, Pa. In all twelve courses of instruction are offered. The list of graduates within the last thirty-seven years numbers 1,248. Several prize scholarships are offered, and provision is made for worthy and needy students whereby they may postpone payment of tuition until after graduation.